I. C. S.
REFERENCE LIBRARY

A SERIES OF TEXTBOOKS PREPARED FOR THE STUDENTS OF THE INTERNATIONAL CORRESPONDENCE SCHOOLS AND CONTAINING IN PERMANENT FORM THE INSTRUCTION PAPERS, EXAMINATION QUESTIONS, AND KEYS USED IN THEIR VARIOUS COURSES

THE STEAM ENGINE
THE INDICATOR
ENGINE TESTING
GOVERNORS
VALVE GEARS
CONDENSERS
COMPOUND ENGINES
ENGINE MANAGEMENT
ENGINE INSTALLATION
PUMPS

SCRANTON
INTERNATIONAL TEXTBOOK COMPANY
10B

PRESS OF
INTERNATIONAL TEXTBOOK COMPANY
SCRANTON, PA.

71171

10B

PREFACE

I.C.S. Reference Library volumes are sent to each student of our regular courses of instruction when he is entitled thereto. These volumes are composed of Sections containing the same text as the Instruction Papers. Each volume contains the Examination Questions pertaining to the text therein, and if a Key to the questions is necessary it is also included. In the case of Courses comprising drawing, the examination work consists of drawing exercises, and the volumes contain no Examination Questions nor Keys. In the back of each volume is a complete index to the contents.

In most cases the contents of the volumes are the same as those of the Instruction Papers. The occasional differences are due to frequent revisions and to the fact that the same volume is sometimes used in more than one Course. The student should send answers to only the Examination Questions in the Instruction Papers comprising his Course.

INTERNATIONAL CORRESPONDENCE SCHOOLS

CONTENTS

THE STEAM ENGINE.

INTRODUCTION.

CLASSIFICATION OF STEAM ENGINES.

1. The great number of the types of the steam engine may be classified as follows:

1. According to the kind of service, as *stationary*, *locomotive*, *marine*, *etc.*

2. According to number and arrangement of cylinders, as *simple*, *compound*, *triple expansion*, *quadruple expansion*, *duplex*, *etc.*

3. According to the type of valve used to control the distribution of steam, as *plain slide valve*, *automatic cut-off*, *Corliss*, *etc.*

4. According to the motion of the piston, as *reciprocating*, *rotary*.

Each of these types may be horizontal or vertical, condensing or non-condensing, and, except the rotary engine, single-acting or double-acting.

2. All the different types of reciprocating engines involve essentially the same principles, and therefore the description of a single type will be sufficient to give a general knowledge of these principles. For this purpose we shall choose the simple slide-valve engine, which is the engine in most common use.

THE RECIPROCATING STEAM ENGINE.

3. Steam may do work by acting on a piston working in a cylinder so as to lift weights or overcome the pressure of the atmosphere; in most cases, however, the work can best be done through the action of shafts and wheels having a continuous rotary motion; it is therefore essential that some

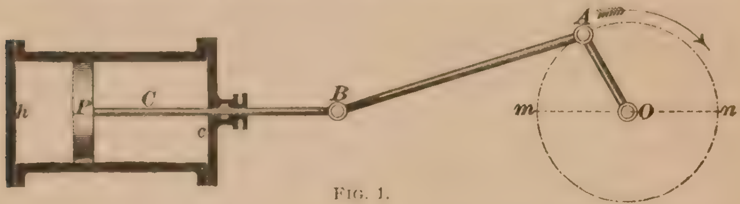


FIG. 1.

method of changing the to-and-fro, or reciprocating, motion of the piston into a continuous rotary motion should be devised. The form of mechanism used for this purpose in practically all types of reciprocating engines is shown in diagrammatic form in Fig. 1.

The steam from the boiler enters one end—say, in this case, the end *h*—of the **cylinder**, and pushes the piston to the other end. By means of another mechanism, called the **valve**, the steam is now admitted to the end *c* of the cylinder, while the end *h* is at the same time allowed to communicate with the atmosphere or with a condenser. The steam in *h* escapes, while that in *c* pushes back the piston to its original position, whence the same operation is repeated.

Attached to the piston, and forming a part of it, is the piston rod *C B*; to the end of *C B* is fastened, by a joint, one end of the link *B A*. The other end of *B A* is joined to the link *A O*; and the other end of *A O* terminates in a shaft *O*, which is fixed in stationary bearings. It is evident that the end of *B A* which is attached to *C B* can move only in a straight line; and since the shaft *O* can only rotate in its bearings, the end of *A O* which is attached to *B A* can move only in a circle.

4. When the piston is at one extreme end of the cylinder, say at *h*, the joint *A* is at the point *m*, and all three

links, $A\ O$, $B\ A$, and $C\ B$, lie in a straight line. As the piston moves to the right, the link $C\ B$ moves also to the right, while the joint A must move in a semicircle $m\ n$. When P arrives at the other end of the cylinder, the joint A is at n, and again $A\ O$, $B\ A$, and $C\ B$ are in a straight line. The piston now moves back to the end h of the cylinder, the joint A moving in the other semicircle from n to m.

The link $A\ O$ is called the **crank,** $B\ A$ the **connecting-rod,** and $C\ B$ the **piston rod.** Those parts that have a to-and-fro, or reciprocating, motion are called the **reciprocating parts.**

The end h of the cylinder is called the **head end,** and the end c the **crank end.** The distance passed over by the piston during half a revolution of the crank is called the **stroke,** and is plainly equal to the diameter of the circle described by the end of the crank—that is, the distance $m\ n$.

5. The engine may run in the direction shown by the arrow in the figure or it may run in the reverse direction. In the former case it is said to *run over* and in the latter case to *run under*.

The stroke from the head end to the crank end of the cylinder, that is, from left to right in the figure, is called the **forward stroke;** the one from crank end to head end, the **return stroke.**

The above simple mechanism perfectly fulfils the office of giving a continuous rotary motion in one direction. A pulley is keyed to the shaft O and the power is transferred by belting from the pulley to shafting or directly to the machinery to be run.

CONSTRUCTION OF A PLAIN SLIDE-VALVE ENGINE.

6. In Fig. 2, the construction of a **plain slide-valve engine** is shown, and in Fig. 3 is shown a section of a steam cylinder. Referring to these figures, H is the head end and C the crank end of the steam cylinder; B and B' are the steam ports; D is the steam chest; E is the exhaust

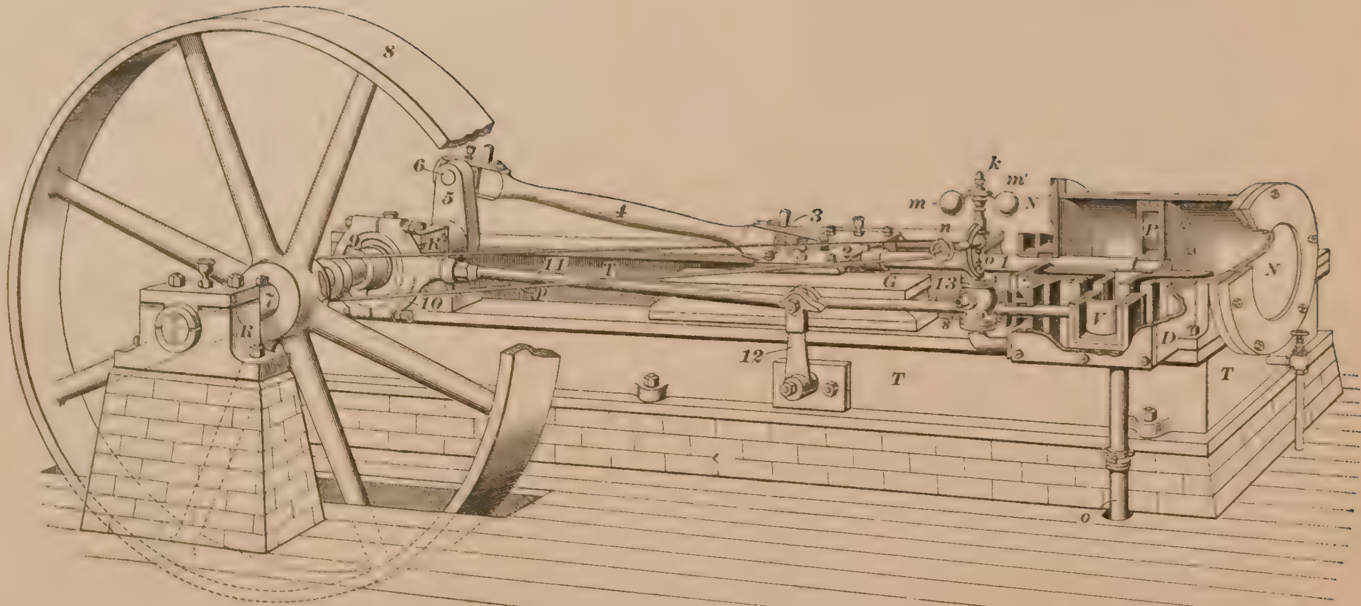


FIG. 2.

port; N and N' are the cylinder heads; S is the steam-supply pipe; O is the exhaust pipe, and connects with the exhaust port E; G is one of the two guide bars; R and R' are the shaft bearings; and T is the bed, or frame, of the engine. The above are all **stationary parts** of the engine, or parts that do not change their relative positions when the engine is in motion. P is the piston; *1* is the piston rod; *2* is the crosshead; *3* is the crosshead pin, which is often called the wristpin; *4* is the connecting-rod; *5* is the crank; *6* is

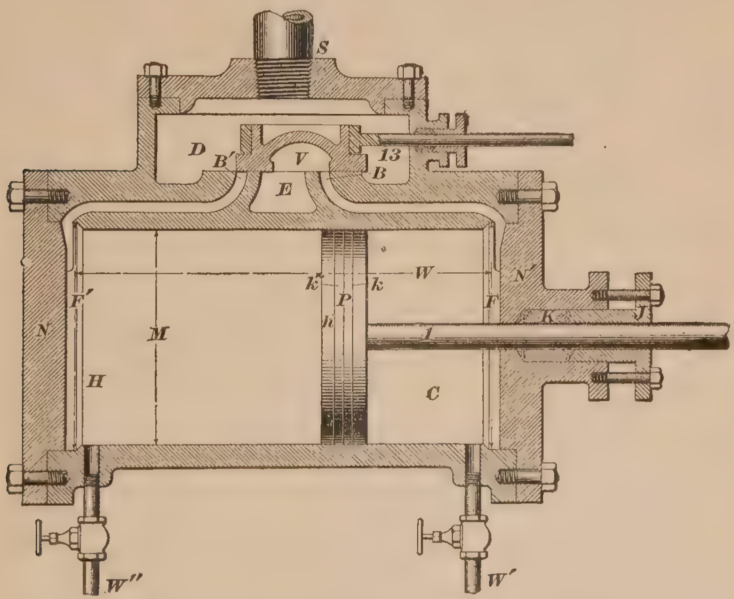


FIG. 3.

the crankpin; *7* is the crank-shaft; *8* is the flywheel; *9* is the eccentric; *10* is the eccentric strap; *11* is the eccentric rod; *12* is the rocker; *13* is the valve rod or stem; and V is the slide valve. These are all movable parts of the engine, or parts that change their relative positions when the engine is in motion. The eccentric, eccentric strap, eccentric rod, rocker, valve stem, and slide valve, which form the mechanism by means of which the steam is distributed, when considered as a whole, are termed the **valve gear** of the engine.

7. The working length of the cylinder is shown by the dimension line *W*. It is slightly less than the distance between the cylinder heads, since a small space must be left between the head and the piston, when the latter is at the end of its stroke. The stroke of the engine is the travel of the piston *P*; since the piston and crosshead are rigidly fastened to the same rod, the stroke must also be equal to the travel of the crosshead. It was shown in Fig. 1 that the stroke is also equal to the diameter of the circle described by the crankpin *6*, or, what is the same thing, it is equal to twice the length of the crank *5*, this length being measured from the center of the crankpin *6* to the center of the crankshaft *7*. The diameter or bore of the cylinder is represented by *M*.

8. The **size** of an engine is generally expressed by giving the diameter of the cylinder and the stroke in inches. Thus, an engine having a cylinder diameter of 16 inches and a stroke of 22 inches is called a $16'' \times 22''$ engine.

9. At the ends *F* and *F'* the cylinder is *counterbored*—that is, for a short distance the bore is greater than *M*. The piston projects partly into this counterbore at the end of each stroke. Were it not for the counterbore, the piston would not wear the cylinder walls their entire length, and shoulders would be formed at each end of the cylinder. When the wear of the joints in the connecting-rod is taken up, the length of the connecting-rod is slightly decreased, and the piston is moved forward slightly towards the crank end of the cylinder. In this case, the shoulder would cause an undesirable pounding of the piston.

Drain cocks *W'* and *W''*, Fig. 3, are fitted to each end of the cylinder, through which any condensed steam may be discharged.

10. The piston fits loosely in the cylinder and has split rings *k* and *k'* inserted, which spring out so as to press against the walls of the cylinder and prevent leakage of steam between the wall of the cylinder and piston. Pistons

are usually supplied with a follower plate *h*, which is bolted to the head end of the piston *P* in order to hold these split rings *k* and *k'* in place. The piston rod *1* is a round bar rigidly connected to both the piston *P* and the crosshead *2*.

11. A stuffingbox *K* in which packing is placed is fitted with a gland *J*, which, when bolted down, compresses the packing around the piston rod *1* and makes a steam-tight joint. This packing is usually made in the form of split rings, which are so placed that the split of the first ring is covered by the solid part of the next ring. When repacking, care should be taken not to cause unnecessary friction by too much pressure from the gland. The guide bars, as *G*, Fig. 2, constrain the crosshead *2* to move exactly in line with the axis of the cylinder, thus relieving the piston rod of all bending stresses.

The connecting-rod *4* forms the connecting link between the crosshead and crank *5*. The joint between crosshead *2* and connecting-rod *4* is made by the crosshead pin *3*, and that between the connecting-rod and crank by the crank-pin *6*. Connecting-rods are usually made from 4 to 6 times the length of the crank, or from 4 to 6 "cranks" in length.

THE ECCENTRIC.

12. Fig. 4 shows the eccentric that imparts motion to the slide valve *V* in Figs. 2 and 3. It consists of a circular disk of iron *a*, which is keyed or fastened by setscrews to the shaft and revolves with it. The center of this disk, which is called the **eccentric sheave,** is at *O*. It is evident that, as the shaft revolves, the center *O* of the sheave *a* will describe the dotted circle *b*, whose center is the center of the shaft. Consequently, the eccentric strap *c* and the eccentric rod *d*, to which it is fastened, will be moved horizontally, during a half revolution, a distance equal to the diameter *e* of the dotted circle. This distance *e* is commonly called the **throw** of the eccentric. The distance *O Q* between the center of the eccentric and center of the shaft is called the **radius** of the eccentric or the **eccentricity.** It is plain

that the throw is *twice* the radius. Attention is here called to the fact that practice varies somewhat in the definition of

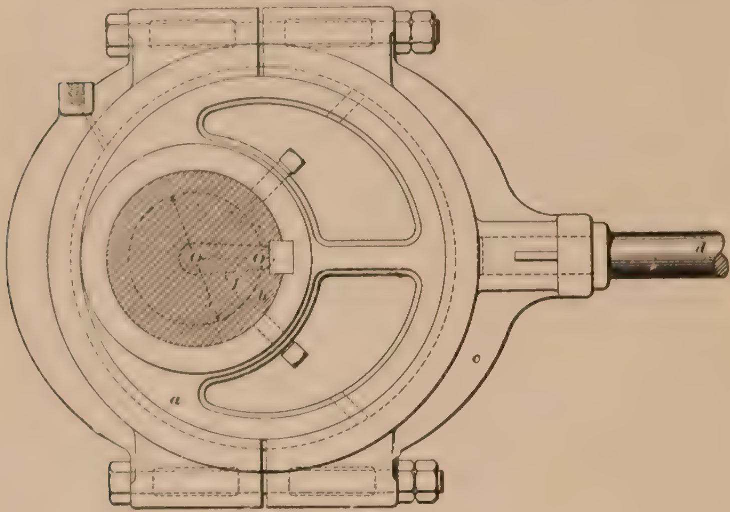


FIG. 4.

the term *throw*. Some engineers call the radius the throw, but by far the greater part define throw as here given.

13. The eccentric is equivalent to a crank whose length is equal to the radius of the eccentric. Thus, if the end of the eccentric rod *d* were attached at *O* to the crank *f* (shown in dotted lines), the crank would give the same motion to the rod that the eccentric does. In plain slide-valve engines, the eccentric is usually keyed to the shaft after being properly adjusted.

The connection between the eccentric rod *11*, Fig. 2, and the valve stem *13* is accomplished in a variety of ways. In Fig. 2, a rocker-arm *12* is used to support the joint between the eccentric rod *11* and the valve stem *13*. The latter must be supported in some manner to prevent its binding in its stuffingbox.

14. Motion of Eccentric and Valve.—As the motion of the valve is given by the eccentric, the valve is in

mid-position in a horizontal engine when the radius of the eccentric is in a vertical position. When QO, Fig. 4, lies horizontally on the right side of Q, the valve V (see Fig. 2) is in its position nearest the head end of the steam chest, and when OQ lies horizontally on the left side of Q, the valve is at the end of its stroke towards the crank end of the steam chest.

THE D SLIDE VALVE AND STEAM DISTRIBUTION.

ACTION OF SLIDE VALVE.

15. Description of the Slide Valve.—Of the different kind of valves used to distribute the steam in the engine cylinder, the **D** slide valve is the most common. A section of such a valve is shown in Fig. 5 in its central position;

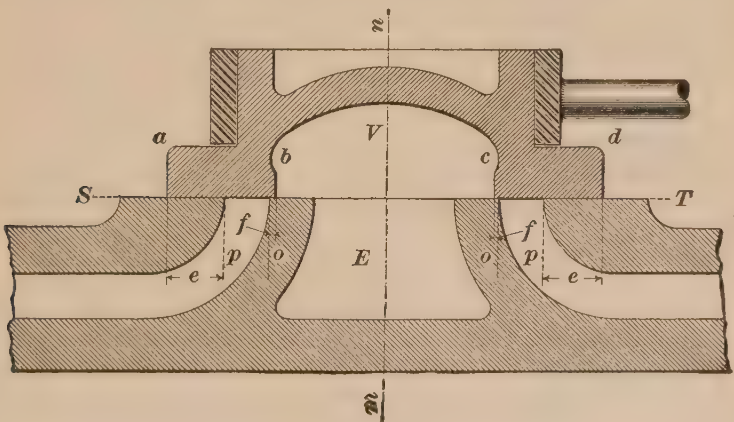


FIG. 5.

p, p are the **steam ports**, o, o the **bridges**, E the **exhaust port**, ST the **valve seat.** The flanges of the valve, ab and cd, are seen to be wider than the ports that they cover. Of this extra width, the parts e, e are called the

outside lap, and the parts f, f the **inside lap.** The valve is here shown in mid-position, i. e., the center line n of the valve coincides with the center line m of the exhaust port.

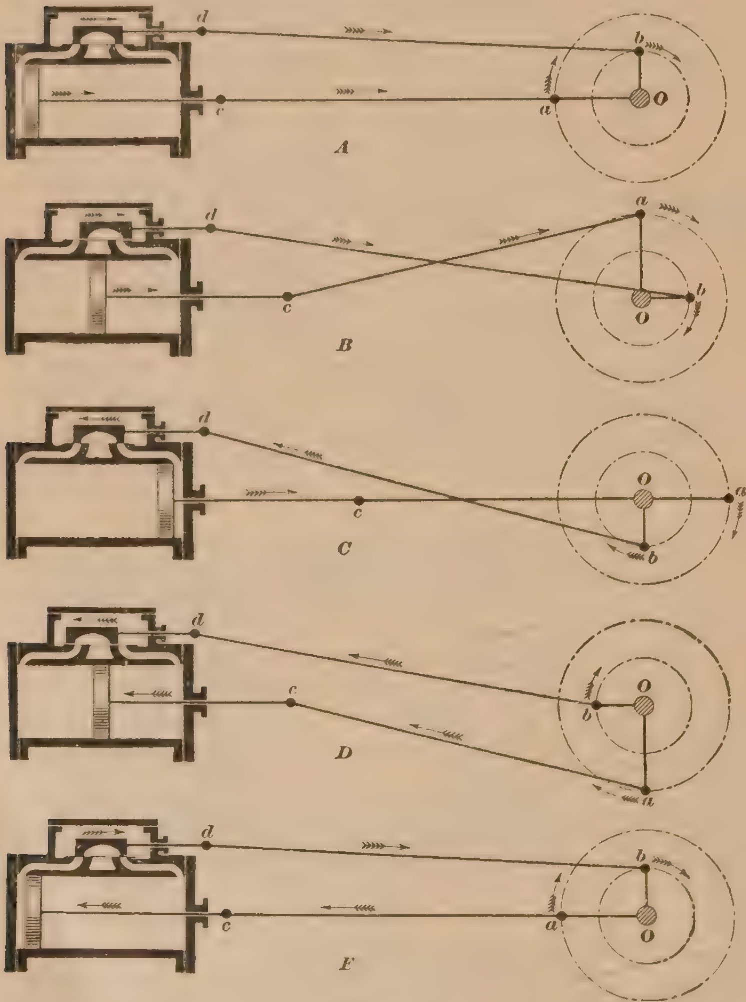


FIG. 6.

16. Action of Valve Without Lap or Lead.—Fig. 6 shows five diagrams that represent a D slide valve without lap

or lead. $O a$ represents the crank; $O b$ the eccentric, which was shown to be equivalent to a crank; $a c$ the connecting-rod; and $b d$ the eccentric rod. It should be remembered that the relative sizes of some of the parts have been greatly exaggerated, particularly the radius of the eccentric circle and the amount of clearance. Diagram A, Fig. 6, represents the piston just on the point of beginning the forward stroke. The valve is moving in the direction of the arrow, and the outer edge is just about to admit steam to the left-hand port. As will be seen, the valve is in its central position, and, consequently, the line joining the center of the shaft and the center of the eccentric (this line will hereafter be called the **eccentric radius**) is vertical. All the parts are about to move in the direction of the arrows. Diagram B shows the positions of the parts when the crank has moved through 90° from its position in A. The piston is at the middle of its stroke, or very nearly there. It would be exactly at the middle of its stroke but for the fact that the connecting-rod makes an angle with the horizontal. It will be assumed here that this has no effect on the position of the piston. The valve has reached the extreme limit of its travel to the right and the eccentric radius $O b$ is horizontal. The left steam port is fully opened for the live steam and the right steam port is fully opened for the exhaust. Another crank movement of 90° places the different parts as shown in diagram C. The piston has reached the end of its forward stroke; the valve is in its central position moving towards the left, and having just closed the left steam port and the right exhaust port, is about to open the right port for the admission of live steam and the left port for the release of exhaust steam. The piston has now traveled one full stroke. Diagram D shows the piston in its central position on the return stroke. The crank is in the position $O a$; the eccentric is horizontal, as represented by $O b$, and the valve is at the farthest point of its travel to the left, the right port being fully open for live steam and the left port fully open for exhaust. In the diagram E the piston has reached the extreme point of the return stroke, the

piston rod, connecting-rod, and crank being all in one straight line; this also occurs in diagrams *A* (which is the same as *E*) and *C*. The valve has been moving to the right and is now in its central position, just on the point of admitting steam to the left port.

These diagrams show that, with no lap or lead, the steam is admitted to the cylinder for the full stroke of the engine; consequently, there can be no cut-off, and therefore no expansion of steam.

The following conclusion is now evident: *With an ordinary* D *slide valve, operated by an eccentric, there can be no cut-off, and therefore no expansion of steam, unless the valve has outside lap.*

17. Lap and Angle of Advance.—The effect of lap on the relative movement of the valve and piston, and also on the movement of the eccentric and crank, is clearly shown in Figs. 7 to 14. In these figures, the valve has both outside and inside lap, but no lead. These diagrams have been distorted, as was done in Fig. 6, in order that the eccentric radius might be long enough to show up well. In Figs. 7 to 14 the eccentric radius is three times as long as it should be for the amount of valve movement shown by the figure. The diameter of the crank circle is also a little greater than the stroke of the piston, for the same reason.

18. Diagram of Pressures in Cylinder.—In order to show the distribution of steam by the valve, a diagram has been drawn above and below each cylinder, those above being marked *M* and those below *N*. These diagrams are supposed to be drawn in the following manner: Imagine it to be possible to connect two small pipes to the piston, one on each side. Suppose each pipe to have a steam-tight piston working in it, the lower side of the pistons being subjected to the steam pressure in the cylinder and the upper side to the atmospheric pressure. Suppose, further, that there is a coiled spring on top of the piston; that a piston rod passes through the center of the spring; and that a pencil is

attached to the end of the piston rod. If a pressure of 10 pounds is required to compress the spring 1 inch, it is evident that for every 10 pounds pressure in the cylinder, the pencil will move upwards 1 inch, and if it touched a sheet of paper, would mark a line on that paper. It will now be presumed that an arrangement like that just described is attached to the steam-engine piston, and that the pencil touches a sheet of paper that is held stationary. Then, when the steam-engine piston moves ahead, the pencil will make straight lines at heights corresponding to the steam pressure on the under sides of the little pistons, except when the pressure of the steam in the cylinder varies, in which case the pencil will move up or down, according as the pressure increases or diminishes.

19. Having made these suppositions clear, let QX, Figs. 7 to 14, represent the line that the pencil would trace if there were a perfect vacuum in the cylinder; i. e., QX is the line of no pressure; also let AB represent the line the pencil would trace if the pressure in the cylinder was just equal to that of the atmosphere, and QY the line of no volume. Then the point Q represents no volume and no pressure. Finally, let $1D$ represent the volume of clearance.

20. Angle of Advance.—Consider Fig. 7. The piston is represented as just beginning the forward stroke and the valve as just commencing to open the left steam port, both moving in the same direction, as shown by the arrows. If the valve had no outside lap, the position of the eccentric center would be at e, but on account of the lap, the valve had to be moved ahead of its central position in order to bring its edge to the edge of the port. To accomplish this, the eccentric center has been moved from e to b, Ob being the position of the eccentric radius. The angle bOe that the eccentric radius makes with the position it would be in if there were no lap or lead is called the **angle of advance.**

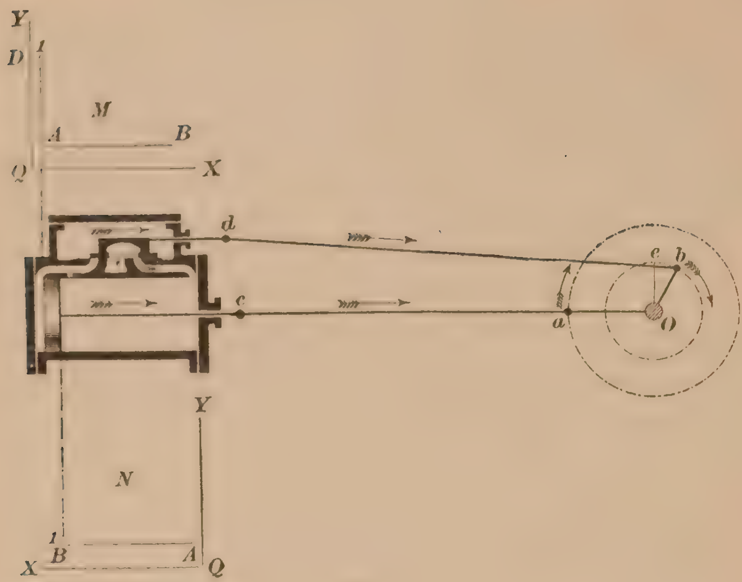


FIG. 7.

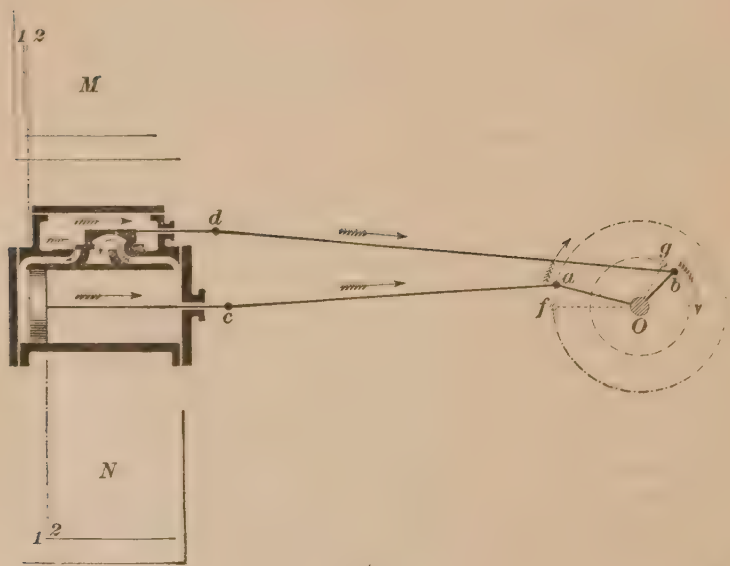


FIG. 8.

21. Back Pressure.—Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance space on the left of the piston, so that the steam acts on the piston at full boiler pressure. If the length of the line *A 1* represents the boiler pressure (gauge), the pencil that registers the pressure on the left side of the piston will be at *1*. The steam on the right side of the piston is flowing (exhausting) into the atmosphere through the exhaust port, as shown by the arrow. As the size of the exhaust port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston in addition to the atmospheric pressure. This is termed **back pressure.** Therefore, in the diagram *N*, let *1* be the position of the second pencil; then, *1 B* is the back pressure.

22. Exhaust Port Fully Open.—Fig. 8 shows the position of the piston and valve when the exhaust port is fully open. The crank has moved from the position *O f* (shown by dotted line) to *O a* and the eccentric center from *g* to *b*. Steam is entering the head end of the cylinder and exhausting at the crank end. The pencils have moved from *1* to *2* on both diagrams *M* and *N*.

23. Valve at the End of Its Stroke.—In Fig. 9, the piston has advanced far enough to enable the valve to reach the end of its stroke and open the port its full width. The crank and eccentric have moved to the positions *O a* and *O b*, the dotted lines showing their last position in Fig. 8. The eccentric radius is horizontal, and any further movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back. In the diagrams *M* and *N*, the pencil has traced the lines *2–3*.

24. Valve on Return Stroke, Steam Port Partly Closed.—Fig. 10 shows the piston still farther advanced on its stroke and the valve as having its inside edge in line with the outside edge of the exhaust port. The left end of

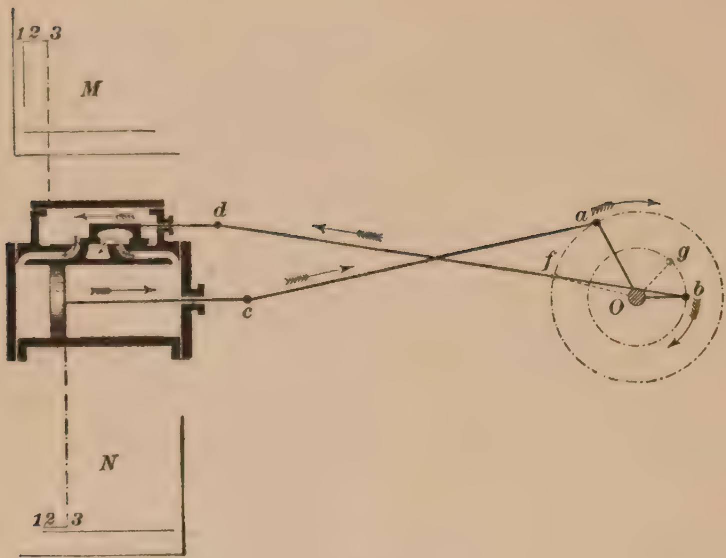


FIG. 9.

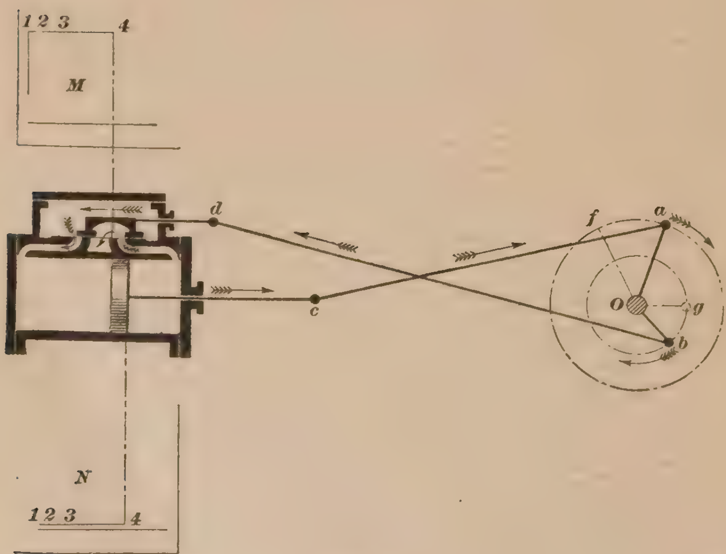


FIG. 10.

the valve has partially closed the steam port. The amount of advancement of the crank and eccentric from their last positions is shown by their distances from the dotted lines. The pencils have traced the lines *3–4* on the diagrams *M* and *N* during this movement of the piston from the last position.

25. Point of Cut-Off.—Fig. 11 marks one of the most important points of the stroke. Here the valve has closed

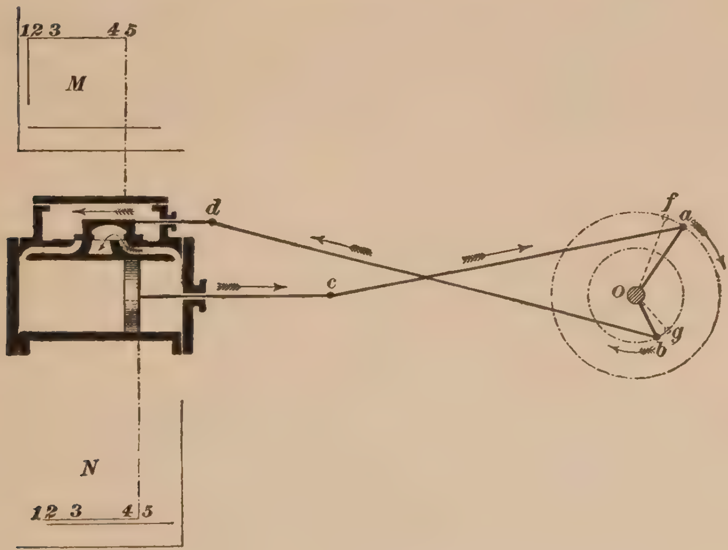


FIG. 11.

the steam port, i. e., cut off the steam, and from here to the end of the stroke the steam in the cylinder expands. This is called the **point of cut-off.** The exhaust port is now partially closed. The crank and eccentric have moved through the angles indicated. During this movement, the pencils have traced the lines *4–5*.

26. Point of Compression.—Fig. 12 shows another very important valve position. Here the inside edge of the

valve closes the exhaust port, and from now to the end of the stroke, the steam in front of the piston is compressed. This point in the stroke is called the **point of compression.**

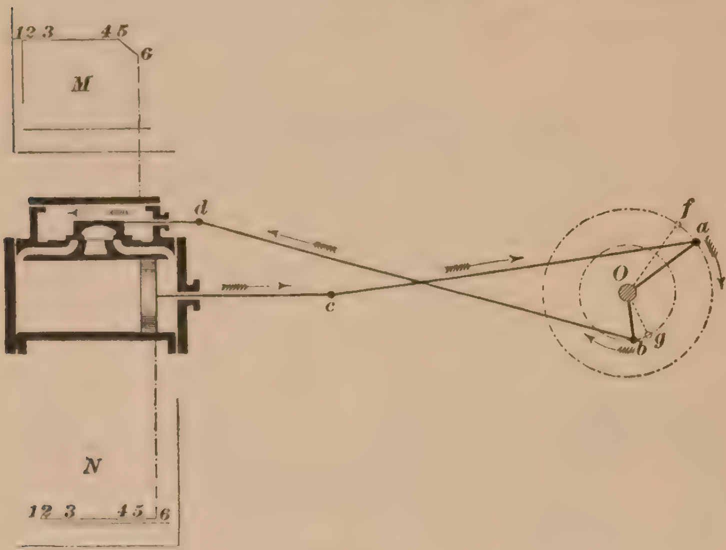


FIG. 12.

In the diagrams *M* and *N*, the lines *5–6* are traced by the pencils. The line *5–6* on the diagram *M* is an expansion line, the pressure falling as the piston moves ahead.

27. Point of Release.—In Fig. 13, the piston has advanced far enough to cause the left inside edge of the valve to be in line with the inside edge of the left port. The slightest movement of the valve to the left will open the left port so that the steam in the left end of the cylinder will pass into the exhaust port. This point of the stroke is called the **point of release.** The work done by expansion theoretically ends here, although, on account of the limitation in the size of the ports, there will still be a slight amount of work done by expansion, owing to the inability of the steam to escape instantly. During this last movement of the piston, the pencils trace the lines *6–7* on the

diagrams M and N. On the diagram M the line *6–7* is a continuation of the expansion line *5–6*, while in the dia-

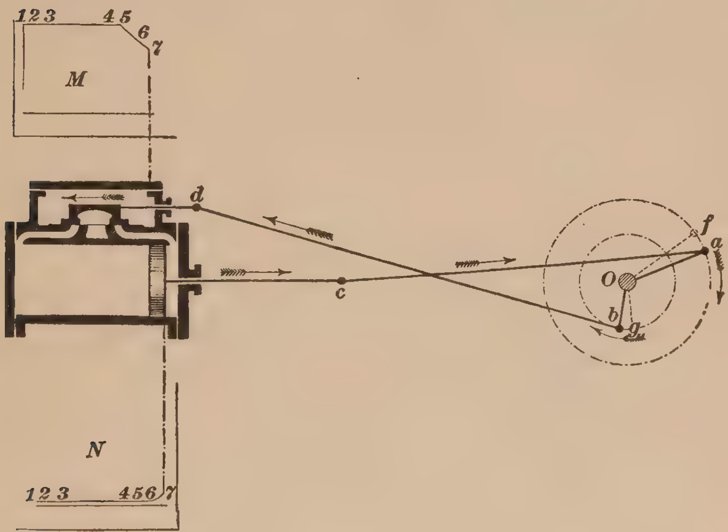


FIG. 13.

gram N it shows part of the compression line, the pressure rapidly increasing as the piston nears the end of the stroke.

28. Piston at End of Stroke.—In Fig. 14, the piston has reached the end of its forward stroke and is about to begin the return stroke. The right outside edge of the valve is in line with the outside edge of the right port. The steam is exhausting from the head end of the cylinder, as shown by the arrows. The crank and eccentric are both diametrically opposite their positions in Fig. 7. In the diagrams M and N, the pencils have traced the lines *7–8*. M shows that the pressure has fallen very rapidly from *7* to *8*, while in N it has risen from *7* to *8*. The very slightest movement of the piston to the left will admit steam to the crank end of the cylinder and cause the pencil to rise to the point *1'*.

During the return stroke the above-described actions of the steam will be repeated, the pencils tracing the dotted lines on the diagrams *M* and *N* in Fig. 14 and the exhaust going through the left port and the steam through the right

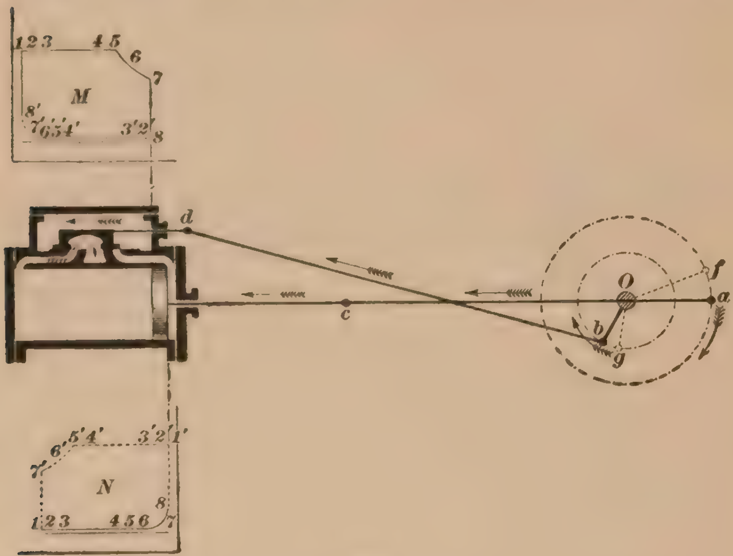


FIG. 14.

port. As the process is so nearly like the preceding, the diagrams have not been drawn, but the student should follow the valve through the different positions and note the effects on the diagrams. To assist him in this, the corresponding points have been numbered as in the foregoing figures.

29. Effects of Lap.—A study of Figs. 7 to 14 should show the effects caused by varying the lap. Thus, in Fig. 11, it is evident that if the outside lap had been less, the valve would not close the left port when its center was in the position shown; consequently, the piston must move farther ahead before the valve can move back far enough to close the port. This, of course, makes the cut-off take place later in the stroke and shortens the expansion. It is likewise

evident that if the valve had more lap, this extra lap would extend beyond the port when the center of the valve was in the position shown. Therefore, the valve would cut off earlier in the stroke and the expansion would be lengthened. Hence, *increasing the outside lap means an earlier cut-off and an increasing expansion*, *while decreasing the outside lap means a later cut-off and a diminished expansion.*

30. Considering the inside lap, it is evident from Fig. 12 that if the inside lap had been less, the exhaust port would not have closed so soon, and consequently the compression would have been later; had the inside lap been greater, the compression would have begun earlier. Fig. 13 shows that with a diminished inside lap, the release would begin earlier, while with an increased inside lap, the release would have taken place later in the stroke. *Increasing the inside lap causes the compression to begin earlier in the stroke and causes the release to take place later. On the other hand, diminishing the inside lap causes the compression to begin later and the release to take place earlier in the stroke.*

31. Lead.—In Fig. 7 the piston is just commencing the forward stroke and the valve is just about to uncover the

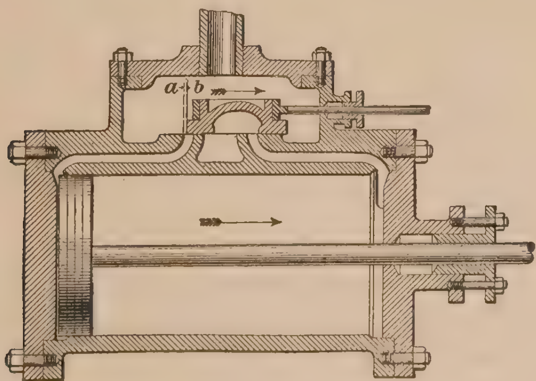


FIG. 15

left steam port. Most engineers, however, prefer to have the port a little open when the piston is at the end of the stroke. That is, the valve, instead of being just at the edge

of the port, as shown in Fig. 7, is moved $\frac{1}{16}$ inch or $\frac{1}{8}$ inch to the right, so that the clearance space is filled with fresh steam before the piston begins its stroke. A valve with lead is shown in Fig. 15. Here the piston is at the end of the stroke and the port is open a distance *a b*. This distance *a b* is the **lead.**

Since, when a valve has lead, it is moved farther to the right than in the position shown in Fig. 7, it is evident that the center *b* of the eccentric must also be moved a little farther to the right, Fig. 7. That is, to give a valve lead, the angle of advance must be increased.

32. Position of the Eccentric.—When the plain slide valve has neither lap nor lead, as in the skeleton diagrams, Fig. 6, it was shown that the eccentric must make an angle of 90° with the crank. Further, when the engine "runs over," as in Fig. 6, the eccentric is *ahead* of the crank. That is, following the direction of the arrows, the eccentric *b* reaches any point on its circle a quarter of a revolution before the crank *a* does. Referring now to Figs. 7 to 14, it is seen that when the valve has lap (or lap and lead), the angle *a O b* between the crank and eccentric is greater than 90°. Following the direction of the arrows, it is seen, however, that the eccentric *b* reaches, say, the lowest point on the circle earlier than the crank *a* reaches the lowest point on its circle. That is, the eccentric is *ahead* of the crank, as in the above case.

Take now the case of an engine that "runs under," as shown in Fig. 16. The crank is in position *a* and is about

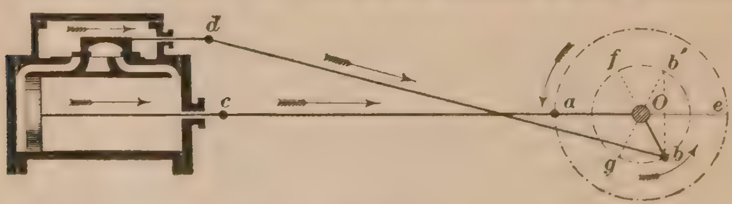


FIG. 16.

to move downwards. Now, the eccentric cannot be in the position *O b′*, for then it would move the valve to the left.

It cannot be opposite, in the position $O\,g$, for in that case, the valve would not be far enough to the right. It must be in the position $O\,b$. An inspection of the diagram shows that, following the direction of the arrows, the eccentric is set *ahead* of the crank, and the angle between the crank and eccentric is $a\,O\,b = 90° +$ the angle of advance.

Hence, for the ordinary slide valve, the following general direction applies: *When the valve rod and eccentric rod move in the same direction, the eccentric is set* **ahead** *of the crank, and the angle between the crank and eccentric is 90° + the angle of advance.* This law is true whether the engine runs "over" or "under."

33. Rocker-Arms.—It frequently happens that the eccentric cannot be so located on the shaft (in the direction of its length) that the eccentric rod and valve stem will be in the same straight line. It can never be done when the valve is on top of the cylinder without inclining the valve seat, now very seldom done, and with the valve on the side of the cylinder, other considerations, such as the location of the flywheel, may interfere. In such cases as this, a lever or rocker-arm may be used.

An example is shown in Fig. 2. It is perfectly evident that when the eccentric rod *11* moves to the left, the valve rod *13* will also move to the left, being compelled to do so by reason of the rocker-arm *12*. It is also plain that the amount of horizontal movement of the valve rod will be the same as it would be if the eccentric were attached directly to the valve rod, thus getting rid of the rocker-arm. The reason for using the rocker-arm in this case is that the eccentric-rod axis and valve-stem axis are not in the same straight line, the eccentric then being thrown too far to the left by the main bearing R'. The valve seat could, in this case, have been placed farther from the center of the cylinder, so as to bring the axes of the two rods in line. This, however, would have made the steam and exhaust ports that much longer. Since it is considered an advantage to have ports as short as possible, a rocker-arm was used.

34. Again, it is sometimes desirable to make the throw of the eccentric less than the valve travel. This may be accomplished by the use of a rocker-arm, as shown in Fig. 17. This rocker is pivoted at g and rotates about that point as a center. The valve rod is joined to the rocker at the end e and the eccentric rod is joined at d, a point between e and g.

Then, the eccentric throw must be smaller than the valve travel by the ratio $g\,d : g\,e \left(= \frac{g\,d}{g\,e}\right)$. For example, suppose the valve travel to be 4 inches, the distance $g\,d = 12$ inches, and $g\,e = 15$ inches. Then, the throw of the eccentric $= 4 \text{ inches} \times \frac{g\,d}{g\,e} = 4 \times \frac{12}{15} = 3.2$ inches.

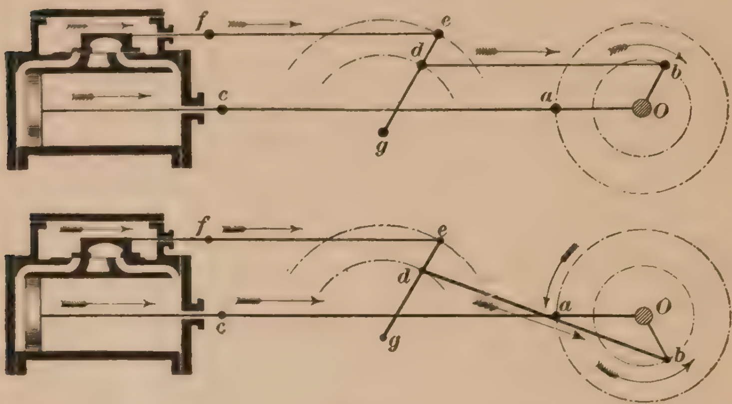


FIG. 17.

When the rocker is arranged as in Fig. 17, whether the engine runs over, as in the upper figure, or under, as in the lower figure, the valve rod and eccentric rod move in the same direction. Consequently, by the direction previously given, the eccentric is set $90° +$ angle of advance *ahead* of the crank.

35. It is often convenient to pivot the rocker near the center, as shown in Fig. 18. Here the points c and d, where

the valve and eccentric rods, respectively, join the rocker, lie on opposite sides of the pivot g. As before, we have the

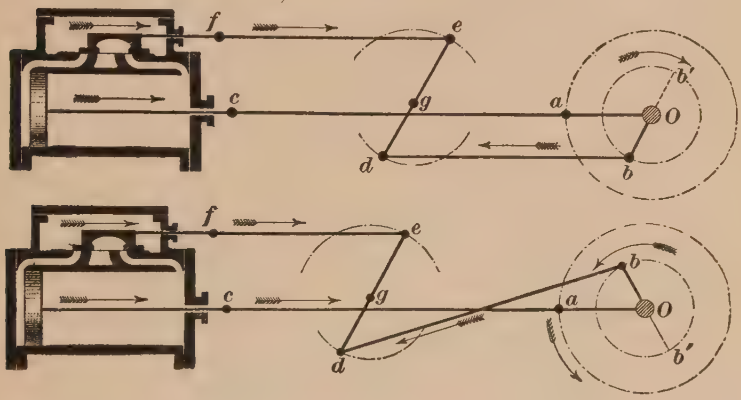


FIG. 18.

proportion — throw of eccentric : valve travel $= gd : ge$, or throw of eccentric $=$ valve travel $\times \frac{gd}{ge}$.

It is easily seen, however, that when the rocker is pivoted near the center, as in Fig. 18, the valve rod and eccentric rod move in opposite directions. Consequently, to give the valve the proper motion, the eccentric rod must at all times move in a direction exactly opposite the direction of a rod attached as shown in Fig. 17. This can only be accomplished by placing the eccentric exactly opposite the position shown in Fig. 17. That is, instead of placing the eccentric $90° +$ angle of advance ahead of the crank, it must be placed $90° -$ angle of advance behind the crank. We have, therefore, the general direction: *In the case of the plain slide valve, if a rocker is so pivoted as to make the valve rod and eccentric rod move in opposite directions, the eccentric must be placed* **behind** *the crank, and the angle between the two is $90° -$ angle of advance.*

36. Direct and Indirect Valves.—A slide valve is said to be **direct** when it opens the left port upon moving to the right and closes it by moving to the left. A valve

is said to be **indirect** when it opens the left steam port by moving to the left and closes it by moving to the right.

The plain slide valve already described is a direct valve. It opens the left port by moving to the right, admits steam past the outside edge, and exhausts it past the inside edge.

37. The **piston valve** shown in Fig. 19 is an example of an indirect valve. It consists of a hollow cylinder sliding

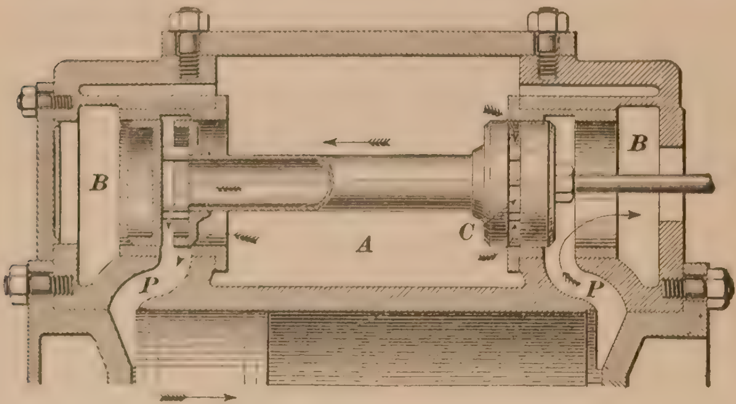


FIG. 19.

in a cylindrical valve seat. The ports *P*, *P* extend clear around the valve. The steam is admitted into the central chamber *A* and the exhaust steam escapes out of the two ends *B*, *B*. As shown in the figure, the piston is just about to start to the right and the valve is moving to the left, thereby uncovering the left steam port and allowing the steam to enter past its inside edge. The valve is, therefore, indirect. To give a larger admission, steam also passes into the center of the valve through the channel *C* and thence into the left port. The exhaust steam meanwhile escapes directly through the right steam port into the chamber *B*.

Attention is called to the fact that a piston valve is not necessarily an *indirect* valve; piston valves are often made as *direct* valves. In the latter case their action is exactly the same as that of the ordinary plain slide valve.

38. Eccentric Positions With Indirect Valves.—It is plain that the direction of motion of an indirect valve is precisely opposite that of a direct valve. Hence, as before explained, the eccentric must be set exactly opposite the position it would have were a direct valve used. We have, then, the following direction for the position of the eccentric: *When an indirect valve is used, set the eccentric* **behind** *the crank and make the angle between them equal 90° – the angle of advance. If a rocker is used that makes the valve rod and the eccentric rod move in opposite directions, set the eccentric* **ahead** *of the crank and make the angle between them equal to 90° + the angle of advance.* This rules applies whether the engine runs "under" or "over."

39. Table of Eccentric Positions.—The position of the eccentric relative to the crank for both the direct and indirect valves, direct and reversing rocker-arms, is given in the table. A rocker of the character shown in Fig. 17 will be called a *direct* rocker. One that changes the direction of the motion, as in Fig. 18, will be called a *reversing* rocker.

ECCENTRIC POSITIONS.

	Kind of Valve.	Kind of Rocker-Arm.	Angle Between Crank and Eccentric.	Position of Eccentric Relative to Crank.
I	Direct	Direct	90° + angle of advance	Ahead of crank
II	Direct	Reversing	90° – angle of advance	Behind crank
III	Indirect	Direct	90° – angle of advance	Behind crank
IV	Indirect	Reversing	90° + angle of advance	Ahead of crank

The above table may be applied equally well whether the engine runs over or runs under. It is simply necessary to remember that to set the eccentric ahead of the crank is to set it so that it reaches a given point in its revolution before the crank reaches the same point in its revolution. For

example, in Fig. 16, suppose the engine to run under, as shown by the arrow. Then, the eccentric $O\ b$ is set ahead of the crank $O\ a$, because it will reach the line $O\ e$ before the crank will. If the eccentric were in the position $O\,f$, it would be behind the crank, because the crank would reach $O\ e$ first. If, now, the engine should be supposed to "run over," then, if the eccentric were in position $O\ b$ or $O\ g$, it would be behind the crank; if in position $O\ f$ or $O\ b'$, it would be ahead of the crank.

DISTURBANCE OF CUT-OFF BY THE CONNECTING-ROD.

40. In Fig. 20, let $a\ b$ represent the path of the center of the wristpin and $c\ d$ the circle described by the center of

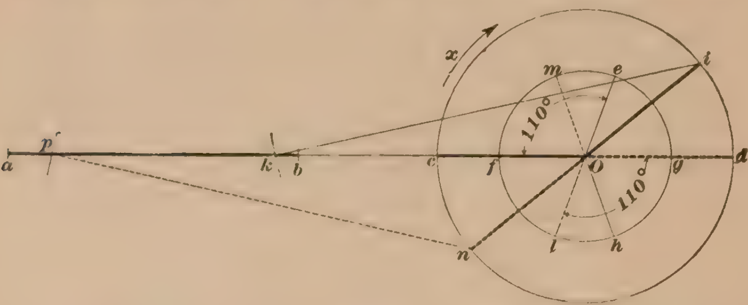


FIG. 20.

the crankpin. Let the diameter of the circle $f\ g$ equal the throw of the eccentric. (This is shown greatly exaggerated.) Assuming the crank to be in the position $O\ c$, that is, on the interior dead center, the length of the line $a\ c$ will represent the length of the connecting-rod. We shall assume that the angle of advance is 20°; further, that the slide valve is set so as not to have any lead. $O\ e$, then, is the position of eccentric when crankpin is at c. Now, let the crankpin move in the direction of the arrow x; that is, let the piston commence its forward stroke. Since the valve has no lead, the slightest movement of the crankpin in the direction of the arrow will cause the valve to open the left steam port. When the eccentric has reached the position $O\ g$, the valve has moved to its farthest position to the right, and any

further movement of the crank will cause the valve to begin to close the steam port. To close the steam port fully, the valve will have to move the same distance to the left that it moved to the right to uncover the port. From this it follows that the eccentric must move through the same angle to close the port that it moved through to open the port. Laying off the angle $g O h = g O e$, $O h$ will represent the position of the eccentric at the time cut-off takes place. Laying off the angle $h O i = c O e$, we find the corresponding crank position. From the point i (the center of the crank-pin) as a center, with a radius equal to the length of the connecting-rod (the length of the line $a c$), describe an arc intersecting the line $a b$ at k; the point k will be the position of the center of the wristpin at the time of cut-off on the forward stroke. When the crank passes the exterior dead center, the right steam port will be opened; and at the moment that the crank occupies the position $O d$, the eccentric will be at $O l$; that is, $90° + 20° = 110°$ ahead of the crank. From what has previously been explained, it will be clear that the cut-off takes place on the return stroke when the eccentric reaches the position $O m$. The corresponding crank position will be $O n$. From n as a center, with a radius equal to the length of the connecting-rod, describe an arc intersecting $a b$ at p, which will give the position of the center of the wristpin at the time of the cut-off on the return stroke.

41. It will be seen at a glance that the cut-off has taken place considerably later on the forward stroke than on the return stroke, since $k b$ is less than $a p$. From this we see that a valve having an equal lap and set so as to have an equal lead cannot cut off equally on the forward and return stroke. If the valve is set so that the cut-off will be equal, the lead will be unequal.

This is due to the use of a connecting-rod. As a general rule, it may be stated that the longer the connecting-rod, the less will be the difference in the points of cut-off; and the shorter the connecting-rod, the greater the difference.

The effects of the connecting-rod on the steam distribution of a simple slide valve may be summarized as follows: *It will cause the valve to cut off and release the steam, as well as close the exhaust port, later on the forward stroke of the piston than on the return stroke.*

FORMS OF SLIDE VALVES.

42. Double-Ported Valves.—The plain **D** slide valve, shown in Fig. 5, is largely used on small engines running

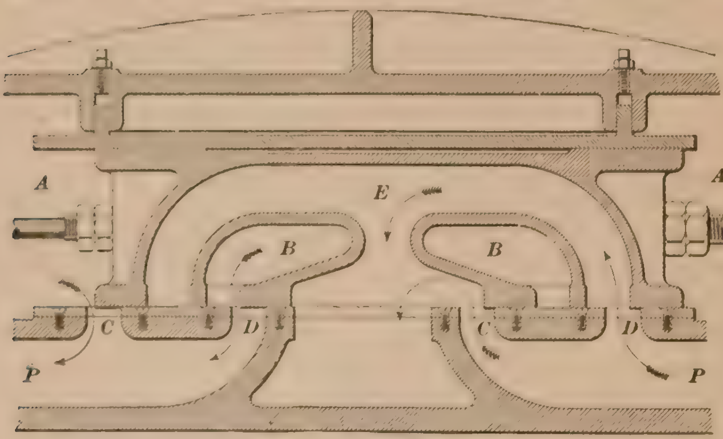


FIG. 21.

at moderately slow speeds. When, however, an engine has a high piston speed, the plain **D** valve does not open the port fast enough to allow the steam to follow up the moving piston and keep up full pressure in the cylinder. To overcome this difficulty, various forms of double-ported valves, one of which is shown in Fig. 21, are used. In Fig. 21, each port *P* has two openings *C* and *D*. The valve is made with two passages *B*, *B* extending through it; these passages connect with the steam chest *A*. In the position shown in the figure, the valve is opening the left steam port. The steam enters the passage *C* past the edge of the valve and enters the passage *D* through the opening in the chamber *B*. In the meantime, the exhaust is escaping from the right

port into the chamber E beneath the valve. It is clear that, with the same travel, the double-ported valve gives double the opening to steam that the plain valve does. Otherwise, the two valves are alike in all respects.

43. The **Allen** or **Trick valve,** shown in Fig. 22, accomplishes the same object. The passage A is cast in the

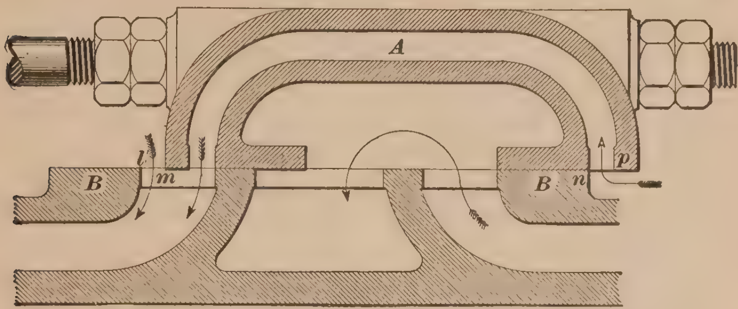


FIG. 22.

valve and extends clear through it. The shoulders B, B of the valve seat are so constructed that when the edge m of the valve is just even with the edge l of the port, the outer edge p of the passage A is just even with the edge n of the shoulder B at the other end of the valve seat. Now, when the valve moves a little to the right, into the position shown in the figure, steam enters the port directly between the edges l and m, as in the case of the ordinary valve. At the same time, the edge p of the passage has moved past the edge n of the valve seat; steam thus enters the passage A and finds there a direct path to the left steam port.

The piston valve shown in Fig. 19 is another example of a valve having a passage through it, by means of which the effective port opening for a given valve travel is doubled.

SETTING THE SLIDE VALVE.

44. Dead Centers.—Referring to Fig. 1, it is plain that when the piston P is at the end of its stroke at the end h of the cylinder, the crankpin A must lie at the point m in the

crankpin circle. In this position the crank OA and connecting-rod AB lie in the same straight line. Likewise, when the piston is at the other end of the stroke, the pin A lies at the point n, and again the crank and connecting-rods are in the same straight line.

When the crank occupies either of these positions, the engine is said to be on its **dead center.** All the pressure of the steam on the piston is transmitted directly to the shaft O, because the reciprocating parts are in a straight line. Consequently, there is no tendency to turn the crank, and the engine cannot be started until turned into a different position. When the crank occupies the position Om, it is said to be on its **inner,** or **head-end,** dead center, and when it occupies the position On, on its **outer,** or **crank-end,** dead center.

45. To Place the Engine on Its Dead Center.—It is sometimes necessary to place the engine exactly on its center

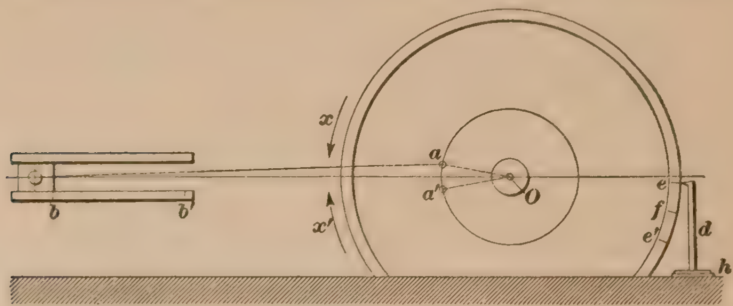


FIG. 23.

in order to set the valve. A common method of doing this is shown in Fig. 23.

When the crosshead is very near the end of its travel, make a mark b on one of the guides opposite the outer edge of the crosshead. Now turn the engine in the direction of the arrow x until the outer edge of the crosshead comes even with the mark b. While the engine is in this position, take a tram d, the length of which is about equal to the distance from the floor to the center of crank-shaft, place

one end upon the floor (or, better, upon a solid block or part of the engine bed), and with the pointed end make a mark e upon the edge of the flywheel. The engine will probably not be exactly on the center; the crankpin will be, say, at a point slightly above the center. Now turn the engine in the direction of the arrow x' until the edge of the crosshead again comes even with the mark. The flywheel will have made nearly a complete revolution, and the crankpin will be at a', the same distance below the center that a was above it. Since the flywheel has made a little less than a full revolution, the mark e on the rim will not now be opposite the marking points of the tram, but the latter will make a new mark e' on the rim. Now, make a mark f half way between the marks e and e', and turn the wheel until the mark f comes opposite the point of the tram. The engine is then exactly on its dead center.

By taking another mark b' at the other end of the guide, the flywheel may be marked for the outer dead center. To insure accuracy, it is well to have both ends of the tram pointed. The lower point then fits into a prick-punch mark, made somewhere in the bed or foundation, and another punch mark on the rim determines the dead-center position.

46. Directions for Setting Slide Valve. — Put the engine on its dead center, place the valve on the seat and connect it with the eccentric rod. Shift the eccentric on the shaft until the valve has the desired lead. Turn the engine in the direction it is to run until it is on the other dead center. If the lead is the same as at the other end, the valve is correctly set; if it is not the same, the valve rod must be lengthened or shortened until the lead is the same at both centers. If, now, the lead is less than desired, shift the eccentric forwards a little on the shaft; if the lead is a little too great, shift the eccentric backwards.

After the valves are set and the engine is started, a pair of indicator diagrams should be taken. The diagrams will show any slight errors in the setting and corrections may be made accordingly.

CLEARANCE: REAL AND APPARENT CUT-OFF.

CLEARANCE.

47. Piston Clearance.—When the crank is on a dead center, the piston is always a short distance from the cylinder head; this allowance is made so that a slight change in the length of the connecting or piston rods will not cause the piston to strike the heads at the end of its stroke. It is also important to have a small space between the piston and head in which any small quantity of water in the cylinder may collect when the piston is at the end of its stroke; the incompressible nature of water would have the effect of breaking some part of the engine if there were no space in which it could collect. The short distance between the piston and the head when the piston is at the end of its stroke is called the **piston clearance.**

48. To measure the piston clearance at either end of the cylinder, first put the engine on its dead center for that end and make a mark on the guides corresponding to some convenient point of the crosshead. Next, disconnect the connecting-rod and push or pull the piston until it strikes the head. The distance of the chosen point on the crosshead from the mark made on the guides is the piston clearance for that end of the stroke.

49. The **clearance volume** or, simply, the **clearance,** is the volume of the space between the piston and cylinder head, when the piston is at the end of its stroke, plus the volume of the port leading to this space. Thus, in Fig. 15, the piston is at the end of its return stroke, and the clearance is the volume of the space between the piston and the left cylinder head plus the volume of the left steam port. In other words, the clearance may be defined as the volume of steam between the valve and the piston when the latter is at the end of the stroke.

50. Measuring the Clearance Volume.—The clearance volume of an engine may be found by putting the

engine on a dead center and pouring in water until the space between the piston and the cylinder head and the volume of the steam port leading into it are filled. The volume of the water poured in is the clearance. Since water is likely to leak past the piston, some engineers advocate the use of a heavy oil for measuring the clearance volume.

51. Method of Expressing Clearance Volume.—The clearance volume may be expressed in cubic feet or cubic inches, but it is more convenient to express it as a percentage of the volume swept through by the piston. For example, suppose the clearance volume of a $12'' \times 18''$ engine is found to be 128 cubic inches. The volume swept through by the piston per stroke is $12^2 \times .7854 \times 18 = 2{,}035.8$ cubic inches. Then, the clearance is $\frac{128 \times 100}{2{,}035.8} = 6.3$ per cent. The clearance may be as low as $\frac{1}{2}$ per cent. in Corliss engines and as high as 14 per cent. in very high-speed engines.

52. Theoretically, there should be no clearance, since the steam that fills the clearance space does no work except during expansion; it is exhausted from the cylinder during the return stroke and represents so much dead loss. This is remedied to some extent by compression. If the compression were carried up to the boiler pressure, there would be very little, if any, loss, since the steam would then fill the entire clearance space at boiler pressure, and the amount of fresh steam needed would be the volume displaced by the piston up to the point of cut-off, the same as if there were no clearance. It is not practicable to build an engine without any clearance, because it is necessary to allow for lost motion and adjustment in the joints of the connecting-rod and because it is also necessary to allow for the formation of water in the cylinder due to the condensation of steam, particularly when starting the engine. As water is practically incompressible, some part of the engine would be broken when the piston reached the end of its stroke, provided there were no clearance space to receive the water; usually

the cylinder heads would be blown off. Neither is it practicable to compress to boiler pressure, as a general rule, for that causes too great strains on the engine. Automatic cut-off, high-speed engines of the best design, with shaft governors, usually compress to about half the boiler pressure and have a clearance of from 5 to 14 per cent. Engines that do not have a high rotative speed, say not over 100 revolutions per minute, have very little compression and very small clearance. Such are the Corliss and other releasing-gear engines.

REAL AND APPARENT CUT-OFF.

53. It is customary, in speaking of the point of cut-off, to say that the engine cuts off at $\frac{1}{2}$ stroke, $\frac{1}{4}$ stroke, etc. By this is meant that the steam is cut off when the piston has completed $\frac{1}{2}$ or $\frac{1}{4}$ of its stroke, as the case may be. For example, if the stroke is 48 inches and the steam is shut off from the cylinder when the piston has moved 18 inches, the cut-off is $\frac{18}{48} = \frac{3}{8}$. The cut-off thus spoken of is the **apparent cut-off.**

54. The **real cut-off** takes account of the clearance space. It is the ratio between the volume of steam in the cylinder and clearance space when the piston is at the cut-off point and the volume of steam in cylinder and clearance space when the piston is at the end of the stroke. For example, let the volume of steam between the valve and piston when the latter is at cut-off be 4 cubic feet. Suppose that when the piston is at the end of its stroke the volume of steam in the cylinder and clearance space is 9 cubic feet. Then, the real cut-off is $\frac{4}{9}$.

55. The **relation between the apparent and real cut-offs** may be shown graphically as follows: Let the

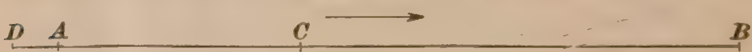


FIG. 24.

length AB, Fig. 24, represent the stroke of the engine. Suppose that the piston is moving in the direction of the

arrow and that the steam is cut off when the piston has reached the point C.

Then, according to the above, the apparent cut-off is $\frac{A\,C}{A\,B}$. It is clear that, since $A\,B$ represents the stroke of the piston, it will also represent, to some scale, the volume swept through by the piston. Now, to the same scale, lay off $A\,D$ equal to the volume of the clearance. Then, from the above definition, the real cut-off is $\frac{D\,C}{D\,B} = \frac{A\,C + A\,D}{A\,B + A\,D}$. Let s represent the apparent cut-off, k the real cut-off, and i the clearance expressed as a per cent. of the stroke. Then, in Fig. 24, $s = \frac{A\,C}{A\,B}$ and $i = \frac{A\,D}{A\,B}$.

Rule 1.—*To find the real cut-off, add the clearance volume, expressed as a per cent. of the stroke, to the apparent cut-off, expressed in per cent., and multiply the sum by 100; divide the product by 100 plus the clearance volume, in per cent.*

Or,
$$k = \frac{(s + i) \times 100}{100 + i}.$$

Example.—In a 13″ × 18″ engine, the steam is cut off when the piston has moved over 8 inches of its stroke. The clearance is 8 per cent. of the volume displaced by the piston. Find the apparent cut-off and real cut-off.

Solution.—The apparent cut-off is $\frac{8 \times 100}{18} = 44.4$ per cent. Applying rule **1**, the real cut-off is found to be $\frac{(44.4 + 8) \times 100}{100 + 8} = 48.5$ per cent. Ans.

56. The **ratio of expansion,** also called the **number of expansions,** is the ratio between the volume of steam in the cylinder and clearance when the piston is at the end of its stroke and the volume in the cylinder and clearance when the piston is at the cut-off point. That is, in Fig. 24, the ratio of expansion is $\frac{D\,B}{D\,C}$. Since $\frac{D\,B}{D\,C} = \frac{1}{\frac{D\,C}{D\,B}} = \frac{1}{k}$, it follows that the ratio of expansion is the reciprocal of the real cut-off. For example, if the volume of steam behind the piston when at the end of its stroke is 15 cubic feet and when at

cut-off is 5 cubic feet, the real cut-off is $\frac{5}{15} = \frac{1}{3}$. The ratio of expansion is $\frac{15}{5} = 3$; in ordinary language, the steam would be said to have 3 expansions.

When the *real* cut-off is given in per cent., the ratio of expansion is found by dividing 100 by the real cut-off in per cent. Thus, if the real cut-off is 25 per cent., the ratio of expansion is $\frac{100}{25} = 4$.

ILLUSTRATIVE EXAMPLE.—Let it be required to find the clearance, the actual cut-off, and the ratio of expansion of a $12'' \times 24''$ engine under the following conditions: When the engine is on its center, the water from a vessel which with the water weighed 5 pounds was poured into the end of the cylinder. After pouring in just enough water to fill the clearance space, the vessel and water were weighed and found to weigh $1\frac{3}{4}$ pounds; consequently, the weight of water poured out of the vessel was $5 - 1\frac{3}{4} = 3\frac{1}{4}$ pounds. The weight of 1 cubic inch of water is .03617 pound. The number of cubic inches poured into the cylinder is, therefore, $\frac{3.25}{.03617} = 89.85$ cubic inches, nearly, which is the volume of the clearance space. The area of the 12-inch piston is $12^2 \times .7854 = 113$ square inches, very nearly; consequently, the piston displacement is $113 \times 24 = 2{,}712$ cubic inches. The clearance volume is, therefore, $\frac{89.85 \times 100}{2{,}712} = 3.31$ per cent. of the piston displacement; in other words, we say that the clearance is 3.31 per cent.

The cut-off takes place when the piston has moved 15 inches of its stroke. The apparent cut-off is, therefore, $\frac{15 \times 100}{24} = 62.5$ per cent. of the stroke. In accordance with rule 1, the real cut-off is $\frac{(62.5 + 3.31) \times 100}{100 + 3.31} = 63.7$ per cent.

In accordance with Art. **56**, the ratio of expansion is $\frac{100}{63.7} = 2$, very nearly.

EXAMPLES FOR PRACTICE.

1. What is (*a*) the clearance volume in cubic inches, and (*b*) the clearance in per cent. of a 16″ × 24″ engine, if 7¼ pounds of water are required to fill the clearance space?

Ans. { (*a*) 200.4 cu. in. (*b*) 4 per cent.

2. If the engine in example 1 cuts off when the piston has made 6 inches of its stroke, what is (*a*) the apparent cut-off, (*b*) the real cut-off, and (*c*) the ratio of expansion?

Ans. { (*a*) ¼, or 25 per cent. (*b*) 27.9 per cent. (*c*) 3.59, nearly.

THE BILGRAM VALVE DIAGRAM.

57. Graphic Method of Determining the Effect of Change in Proportion of Valves.—The action of the valve of a plain slide-valve engine when operated by an eccentric can be readily analyzed by means of a diagram that has been designed by Mr. Hugo Bilgram. This diagram is extremely useful not only in the analysis of an existing slide-valve gear, but as it also exhibits in a graphic form the effects of any change in the proportions of the valve, it is invaluable in the design of a new valve.

The valve diagram and its application to an existing valve gear is shown in Fig. 25. In a case of this kind, the outside and inside lap of the valve, the travel of the valve, and the stroke of the engine are known; and the lead, if not known, may be assumed. With these data, the amount that the steam ports are opened (the port opening), the point of cut-off, the point of release, the point of exhaust closure, and the angle of advance of the eccentric can be determined.

58. To any convenient scale, draw on the line $a\,b$, with o as a center, the semicircle $a\,d\,b$ having a radius equal to that of the crank (one-half the stroke). About o as a center and with a radius equal to one-half the valve travel, describe the semicircle $a'\,o'\,b'$. Draw a line $g\,h$ parallel to $a\,b$ and at a distance from it equal to the lead. With a radius equal to the outside lap of the valve and with a center o' on the semicircle $a'\,o'\,b'$ describe a circle r that is tangent

to $g h$. The position of the center o' is easiest found by trial. About o' as a center and with a radius equal to the inside lap of the valve, describe a circle s. Next draw the straight lines $o d$, $o l$, and $o m$ tangent, respectively, to the outside lap circle r and the inside lap circle s. Through o

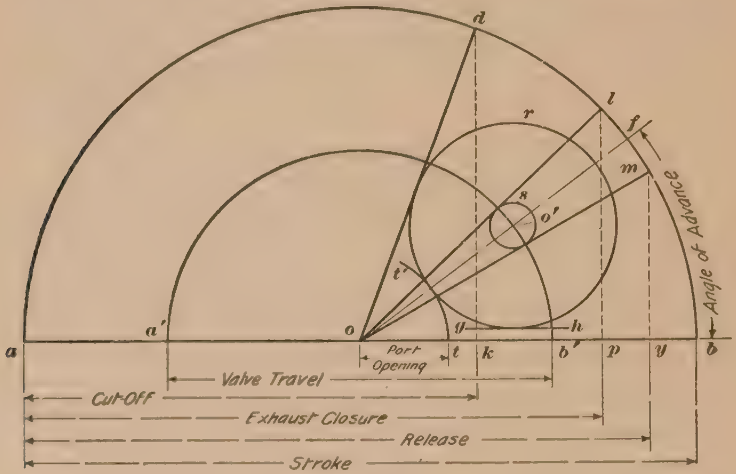


FIG. 25.

and o' draw the straight line $o f$. From the points of intersection d, l, and m of the lines $o d$, $o l$, and $o m$ with the semicircle $a d b$, drop perpendiculars, as $d k$, $l p$, and $m y$ on the straight line $a b$. About o as a center describe an arc tangent to the outside lap circle and intersecting $a b$ in t.

In the diagram just drawn, the distance $o t$ represents the port opening to the same scale to which the diagram was drawn; the distance $a k$ shows the piston movement up to the point of cut-off; the distance $a y$ shows the piston movement up to the point of release, and the distance $a p$ shows the piston movement up to the point where the exhaust port is closed, i. e., up to the point where compression begins. The angle $f o b$ is the angle of advance.

59. When the valve has no inside lap, no inside lap circle can be drawn; in that case, drop a perpendicular from

the intersection point f of the line $o f$ with the semicircle $a d b$ to the line $a b$. The distance between a and the point of intersection of this perpendicular with the line $a b$ will then represent the piston movement up to the points of compression and release.

60. A careful study of the diagram will show the effects of changes in the valve proportions and valve setting. Thus, suppose that, the valve proportions remaining the same, it has been decided to give more lead. Then, the lead line $g h$ being at a greater distance than before from $a b$, the center o' will be higher up and farther to the left; in consequence, the intersection points d, l, and m will also be to the left of their former positions and the intersections of perpendiculars dropped from these points on $a b$ will be nearer a than before. In other words, the increase of lead causes the different events to take place earlier. Since o' will occupy a different position than formerly, the angle $f o b$ (the angle of advance) will now be greater; this shows that, with the valve proportions remaining the same, increasing the lead is impossible without increasing the angle of advance.

61. Suppose that the inside lap is *decreased*, all other valve proportions and the lead remaining the same. Then, since the inside lap circle s will be smaller, the point l will be nearer f than before, and, consequently, the intersection of a perpendicular dropped from l on $a b$ will be to the *right* of p; that is, the exhaust closure will take place later. The intersection point m will also be closer to f, and, obviously, the intersection of a perpendicular dropped from it on $a b$ will be to the *left* of y; that is, the release will take place earlier.

Suppose that the outside lap is *increased* in order to get an earlier cut-off and that the lead is to remain as before. Then, owing to the greater diameter of the circle r, its center o' will be farther to the left, and, consequently, the tangent lines $o d$, $o l$, and $o m$, and their perpendiculars $d k$, $l p$, and $m y$ will also move to the left, thus indicating that all the events will take place earlier.

62. Assume that while the outside lap was increased, no change was made in valve travel. Then, since the lap circle *r* is larger than before, it follows that *o t* will be smaller; that is, an increase of outside lap not accompanied by an increase of the valve travel will cause a decrease in the port opening. From this it is evident that in order to keep the port opening constant, an increase of outside lap must be accompanied by an increase of the valve travel.

By studying in the manner indicated in the preceding articles the effect on the valve diagram of any change in the valve proportions, in the angle of advance, etc., the effects on the steam distribution can be noted easily.

63. In designing a new valve for an engine, the port opening, the point of cut-off, and the stroke of the engine are known; the lead must be fixed upon. With these data, the valve proportions, the valve travel, and the angle of advance are readily determined by means of the Bilgram valve diagram.

On the line *a b* and from *o* as a center, describe the semicircle *a d b* to represent the path of the crankpin, using any convenient scale. Draw the lead line *g h* parallel to *a b* and at a distance equal to the lead from it. With a radius equal to the port opening, describe an arc *t t'* about *o* as a center. On *a b* lay off *a k* equal to the desired cut-off. At *k* erect a perpendicular, and from its point of intersection *d* with the semicircle *a d b* draw the straight line *d o*. Now, by trial, find the radius and the position of the center *o'* of a circle that will be tangent to the lines *o d* and *g h* and tangent to the arc *t t'*. The radius of this circle represents the outside lap required, while the distance of the center *o'* from *o* represents one-half the valve travel. By drawing a straight line, as *o f*, through *o* and *o'*, the angle of advance is determined. The question of how much, if any, inside lap to give is one that each designer must answer for himself, remembering that the giving of inside lap makes release later and exhaust closure earlier.

SIZE OF STEAM PASSAGES.

64. The average practice of steam-engine builders is to proportion the steam passages so that the steam will flow at a velocity of 6,000 feet per minute through the main steam pipe, 6,000 feet per minute through steam ports that are relatively long and tortuous, as the steam ports of plain slide-valve engines, 7,500 feet per minute through very short and direct steam ports, 4,000 feet per minute through the exhaust ports, and also through the exhaust pipe.

65. With these velocities as a basis, the following rules for proportioning the steam passages have been deduced. In the case of steam pipes and exhaust pipes, the commercial size of pipe whose area is nearest the calculated area should be selected. In case either the steam pipe or the exhaust pipe, or both, is very long, say above 200 feet, it may be advisable to select a pipe one size larger than the one whose area is nearest the calculated area. In case the steam pipe is but poorly or not all protected by covering, the use of a larger size of pipe is especially necessary.

66. Rules and Formulas for Calculating the Sizes of Steam Passages.—

Let a = area of steam port in square inches;
b = area of exhaust port and pipe in square inches;
c = area of steam pipe leading to engine;
d = piston speed in feet per minute;
e = area of piston in square inches.

Rule 2.—*To find the area of the steam pipe leading to an engine or pump, multiply the area of the piston by the piston speed in feet per minute and divide the product by 6,000.*

Or, $$c = \frac{de}{6,000}.$$

EXAMPLE.—A 14″ × 36″ engine is to run 100 revolutions per minute. What size should the steam pipe be?

Solution.—The piston speed in feet per minute is $\frac{36}{12} \times 100 \times 2 = 600$ feet. The area of the piston is $14^2 \times .7854 = 153.938$, say 154 square inches. Applying rule 2, we get

$$c = \frac{154 \times 600}{6,000} = 15.4 \text{ square inches.}$$

The nearest commercial size of pipe is $4\frac{1}{2}$ in. nominal diameter, whose internal area is 15.939 sq. in. Ans.

67. Rule 3.—*To find the area of the exhaust pipe or exhaust port for an engine or steam pump, multiply the area of the piston by the piston speed in feet per minute and divide the product by 4,000.*

Or, $$b = \frac{de}{4,000}.$$

Example.—What should be the area of the exhaust port and what should be the size of the exhaust pipe of the engine mentioned in the example given in Art. **66**?

Solution.—Applying the rule just given, we get

$$b = \frac{154 \times 600}{4,000} = 23.1 \text{ square inches}$$

as the area of the exhaust port. Since a 6-inch pipe has an actual internal area of 28.889 square inches, while the next smaller commercial size of pipe, viz., 5-inch, has an area of but 19.99 square inches, experienced engineers would select the 6-inch pipe, in order not to cramp the exhaust. Ans.

68. Rule 4.—*To find the area of the steam port, multiply the area of the piston by the piston speed in feet per minute; divide the product by 7,500 if the port is short and direct, and by 6,000 if the port is long and tortuous.*

Or, $$a = \frac{de}{7,500} \text{ for a short port,}$$

and $$a = \frac{de}{6,000} \text{ for a long port.}$$

Example.—What should be the area of the steam port for the engine given in the example of Art. **66**, if the port is short?

Solution.—Applying rule 4, we get

$$a = \frac{154 \times 600}{7,500} = 12.32 \text{ sq. in. Ans.}$$

THE ROTARY ENGINE.

69. Since the time of Watt, it has been the aim of many inventors to produce an engine in which the piston has a rotary motion, thus dispensing with the connecting-rod and crank. Innumerable designs have been proposed and patented, many of which have been actually tried; except for special service where economy in the use of steam is but a minor consideration, they have all proved commercial failures. It is a very simple matter to design a rotary engine that will turn (run); it is an entirely different matter, however, to have a rotary engine develop in constant and extended service a horsepower on the same steam consumption as a reciprocating steam engine.

70. Rotary engines have been constructed in a great variety of forms, many of which can only be characterized as freaks. The remainder usually consist either of a rotary piston of some suitable form, bearing against a rolling, sliding, or swinging abutment, or a design of interlocking pistons similar to that found in the Root blower. Abutments, no matter how carefully designed, made, and fitted, cannot be kept steam-tight for any length of time, and generally will cause bad steam leakage into the exhaust in a short time; rotary engines of the interlocking piston pattern either commence to leak badly after short service or are very noisy. Some rotary engines are valveless and very simple, but extremely costly to operate. Some have eccentric pistons; these rapidly wear the cylinder walls and bearings owing to the difficulty of counterbalancing them properly for high speeds. On the whole it can be safely stated that the rotary engine, owing to inherent irremovable constructive difficulties, will never be a serious commercial competitor of the reciprocating engine.

THE STEAM TURBINE.

71. Two eminent engineers, Mr. Parsons and Mr. Laval, have developed a steam engine working on an entirely different principle than the ordinary steam engine. Instead of making use of the pressure of steam, they utilize the kinetic energy contained in a mass of steam moving with a very high velocity, jets of steam impinging against the blades or vanes of a wheel fitted inside of a suitable casing and thus rotating the wheel at a high speed. This kind of an engine, from its similarity to the turbine, is called a **steam turbine.** A limited number of steam turbines are in use, the latest designs giving an economy about equal to that of the reciprocating steam engine.

THE INDICATOR.

INDICATORS AND REDUCING MOTIONS.

CONSTRUCTION OF INDICATOR.

GENERAL CONSTRUCTION.

1. The **indicator** is an instrument that can be readily applied to a steam engine for the purpose of obtaining a

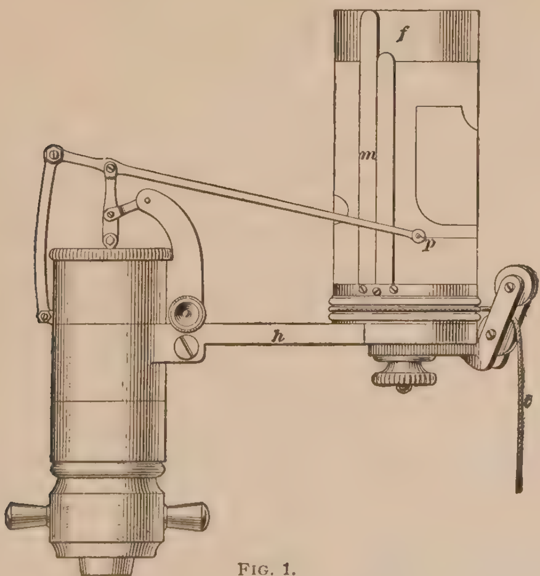


FIG. 1.

diagram of the pressures in the cylinder. It is made in a variety of forms that differ, however, only in minor details;

the general principles involved in all will readily be understood by reference to Fig. 1, which shows the general appearance of an indicator, and Fig. 2, which shows one in

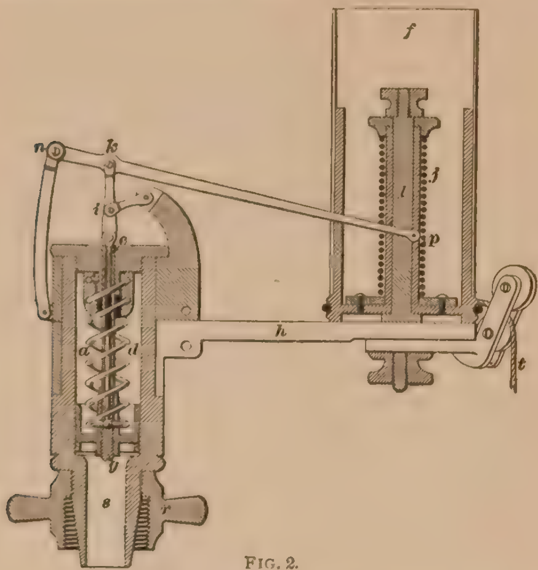


FIG. 2.

section. The instrument consists essentially of a cylinder *a*, Fig. 2, containing the piston *g* and the spring *d*. By turning a cock connected to the small pipe to which the indicator is attached, steam may be admitted to, or shut off from, the cylinder of the indicator at pleasure. When steam is admitted through the channel *s*, its pressure causes the piston *g* to rise. The spring *d* is compressed and resists the upward movement of the piston. The height to which the piston rises should then be in exact proportion to the pressure of the steam, and as the steam pressure rises and falls, the piston must rise and fall accordingly.

2. To register this pressure, a pencil might be attached to the end of the piston rod *e*, the point of the pencil being made to press against a piece of paper. It is desirable, however, to restrict the maximum travel of the piston to about ½ inch, while the height of the diagram may

advantageously be 2 inches or more. To obtain a long pencil movement combined with a short travel of the piston, the pencil is attached at p to the long end of the lever $n\,k\,p$. The fulcrum of the lever is at n. The piston rod is connected to it at k through the link $i\,k$. The pencil motion is thus $\frac{p\,n}{n\,k}$ times the piston travel. This ratio $\frac{p\,n}{n\,k}$ is, for most indicators, either 4, 5, or 6. The point p is forced to move in a vertical straight line by the arrangement of the links and joints i, e, n, and k, forming what is called a **parallel motion.**

DETAILS OF INDICATOR.

3. The Spring.—The height to which the piston will rise under a given steam pressure depends upon the stiffness of the spring. Indicators are usually furnished with a number of springs of varying degrees of stiffness, which are distinguished by the numbers 10, 20, 30, 40, etc. These numbers indicate the pressure per square inch required to raise the pencil 1 inch and are called the **scale of the spring.** Thus, if a 40 spring is used, a pressure of 40 pounds per square inch raises the pencil 1 inch, and the vertical scale of the diagram is, therefore, 40 pounds per inch; that is, the vertical distance in inches of any point on the diagram from the atmospheric line, multiplied by 40, gives the gauge pressure per square inch at that point. The scale of the spring chosen should not be less than one-half the boiler pressure. For example, we would choose a 40 spring for a steam pressure of 75 pounds per square inch.

4. The Paper Drum.—The indicator must not merely register pressures, but it must register them in relation to the position of the piston. To accomplish this object, a cylindrical drum f, Figs. 1 and 2, is provided. This drum can be revolved on its axis l by pulling the cord t, which is coiled around it. When the pull is released, the spring j rotates the drum back to its original position. If, now, the cord t be attached to some part of the engine that has

a motion proportional to the motion of the piston, the motion of the drum will also be proportional to the motion of the piston.

CONNECTING INDICATOR TO ENGINE CYLINDER.

5. To attach the indicator to the engine, a hole is drilled in the clearance space of the cylinder and tapped for a ½-inch nipple. If this hole is in the top of the cylinder, the indicator cock may be screwed directly into it, or, if more convenient, a nipple and coupling may be used. If the cylinder is tapped at the side, a nipple and elbow may be used so as to bring the indicator into a vertical position; since, however, it is desirable to keep the connections to the indicator as short and direct as practicable, some engineers prefer to omit the elbow and attach the indicator in a horizontal position. The indicator is attached directly to the cock by the nut *r*, Fig. 2, which wedges the conical projection *s* of the indicator tightly into the cock and thus prevents leakage of steam. On account of the resistance offered by the pipe and elbows to the flow of steam to the indicator, it is preferable to have an indicator at each end of the cylinder, but if that is not convenient, one indicator may be connected with both ends of the cylinder by means of a three-way cock.

6. Most cylinders of the better class of engines are now provided with bosses having holes tapped in them for the convenient application of the indicator; in many old engines, however, no special provision for the indicator has been made. In such cases care must be taken to drill holes that will not be covered with the piston when it is near the end of the stroke. If the hole cannot be tapped directly into the clearance space, a passage must be chipped to the clearance space in order that the steam can reach the indicator.

7. The indicator connection should never be on the side of the cylinder directly opposite to the steam ports; the current of entering steam would strike against the opening

leading to the indicator, and the pressure shown by the diagram would thereby be considerably increased. With many engines that have not been provided with special indicator attachments, it will be found that holes for this purpose may be conveniently drilled and tapped in the cylinder heads.

INDICATORS FOR SPECIAL PURPOSES.

8. Gas-Engine Indicators. — An indicator specially adapted to gas engines, and one that is sometimes applied to steam engines, pumps, and hydraulic machinery when high pressures are used, is shown in Fig. 3. The cylinder has two bores *a* and *b*. The larger bore *a* is $\frac{1}{2}$ square inch in area (the size usually employed when testing a steam engine), and the area of the smaller bore *b* is $\frac{1}{4}$ square inch. The piston *c* is fitted to the smaller bore and is that used when indicating a gas engine or a very high pressure steam engine. It gives but half the movement of the pencil given by the larger piston used in *a*; so, if the spring used is stamped 40, the calculations, when using the smaller piston, must be made as if an 80-pound spring had been used. The pencil movement is also of special design, the moving parts being stronger and more rigid than those used on regular steam-engine patterns With the regulation piston of $\frac{1}{2}$ square inch area in chamber *a*, the indicator may be used for steam engines with ordinary pressures.

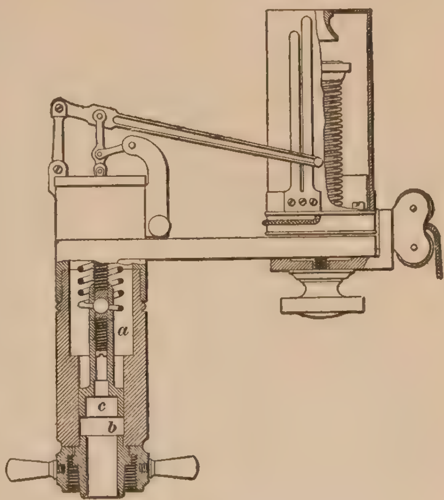


FIG. 3.

9. Ammonia Indicators.—For use on the ammonia cylinders of refrigerating machines, it is preferable to use a special indicator the working parts of which are made of steel instead of brass, because ammonia has no effect on steel, but rapidly corrodes brass. In case it is not possible to procure an ammonia indicator, an ordinary steam-engine indicator will answer the purpose, provided the piston is removed after every set of cards is taken and both cylinder and piston are wiped dry and well covered with oil. This will prevent the ammonia gas from attacking the portions of the indicator made of brass.

REDUCING MOTIONS.

PANTOGRAPH MOTIONS.

10. Purpose of a Reducing Motion.—The motion of the paper drum is nearly always taken from the crosshead. However, since the stroke of the crosshead is longer than the circumference of the drum, it is necessary to arrange a form of mechanism some point of which will copy to a reduced scale the stroke of the piston. Such a mechanism is called a **reducing motion.**

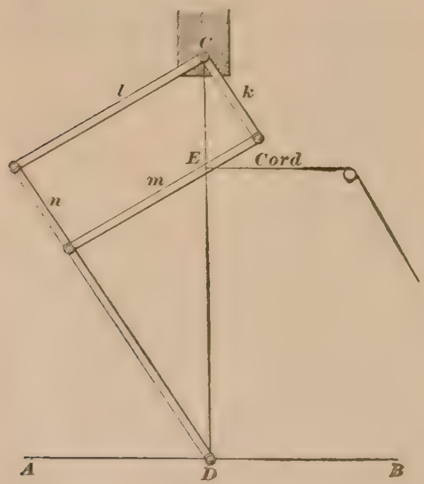


FIG. 4.

11. The **pantograph,** Fig. 4, is an excellent form of reducing motion. It consists of four links joined together in the form of a parallelogram. One of the links n is prolonged and is pivoted at the end to the crosshead D. The opposite corner of the parallelogram

is pivoted to the fixed point C. The cord is attached to the point E on the link m, which point must be on the straight line connecting C and D. $A B$ represents the length of the stroke. Letting h represent the length of the indicator diagram, we have the following proportions: $A B : h = C D : C E$, or $\frac{A B}{h} = \frac{C D}{C E}$.

12. The **lazy tongs,** Fig. 5, is a modified form of the pantograph that is much used as a reducing motion. It

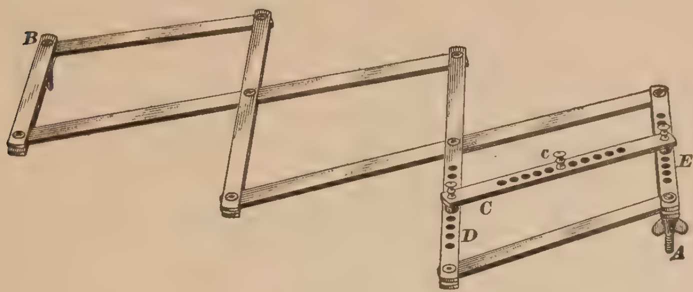


FIG. 5.

consists of a series of bars joined together in such a manner as to form a flexible frame. The joint A of the frame is attached to any convenient stationary point on the engine or its surroundings and B is attached to the crosshead. The bar C is provided with a number of holes, in one of which is placed a pin c to which the indicator cord is attached. The bars D and E to which C is joined are provided with a series of holes, and C may be placed in any position in which its ends are attached to holes similarly located in the two bars. For each position of C, one of the holes in it will be in line with the two joints A and B at the extremity of the frame; this is the hole in which the pin c is to be placed for that position of C.

13. The ratio of the length of the diagram to the length of the stroke is equal to the ratio of the distance $A c$ to the distance $A B$; this is true for any distance between the points A and B. To find the correct position of the bar C

for a given length of card, when the length of stroke is known, set the points A and B at some convenient distance apart; multiply this distance by the desired length of the card and divide the product by the length of the stroke; the quotient so obtained will be the distance from A at which to set the pin c, keeping A and B at the distance apart to which they were set at the beginning of the operation. A very convenient method of locating C is to make the distance $A B$ equal to the length of the stroke and then locate C so that the distance $A c$ is, as nearly as possible, equal to the desired length of the diagram. For example, let it be desired to take a diagram $3\frac{1}{2}$ inches long from an engine having a stroke of 32 inches; make the distance $A B$ 32 inches and then attach C to the holes in D and E that will make the distance $A c$, as near as may be $3\frac{1}{2}$ inches.

14. The pantograph and lazy tongs are theoretically correct reducing motions; that is, the motion imparted to the indicator cord is exactly proportional to the motion of the crosshead. The point to which the cord is attached moves in a straight line parallel to the direction of motion of the crosshead. The fixed point of either the pantograph or lazy tongs may be at any place that will enable the cord to be led to the indicator in the shortest and most direct manner; it is not necessary, as is sometimes assumed, to locate the fixed point on a line equidistant from both ends of the stroke. In locating the point of attachment, however, it is important to guard against striking the joints of the frame at the ends of the stroke; neglect of this precaution may result in breaking the reducing motion. A disadvantage of the pantograph is the danger of lost motion due to wear in the joints.

SWINGING-LEVER REDUCING MOTIONS.

15. A lever with one end pivoted at some convenient fixed point and the other attached to the crosshead of the engine forms one of the simplest reducing motions; and if the device is correctly designed and carefully constructed,

it can be made to give as accurate results as are obtainable in any way. Two common forms of swinging-lever reducing motions are those illustrated in Figs. 6 and 7. The form shown in Fig. 6 is called the **slotted swinging lever.**

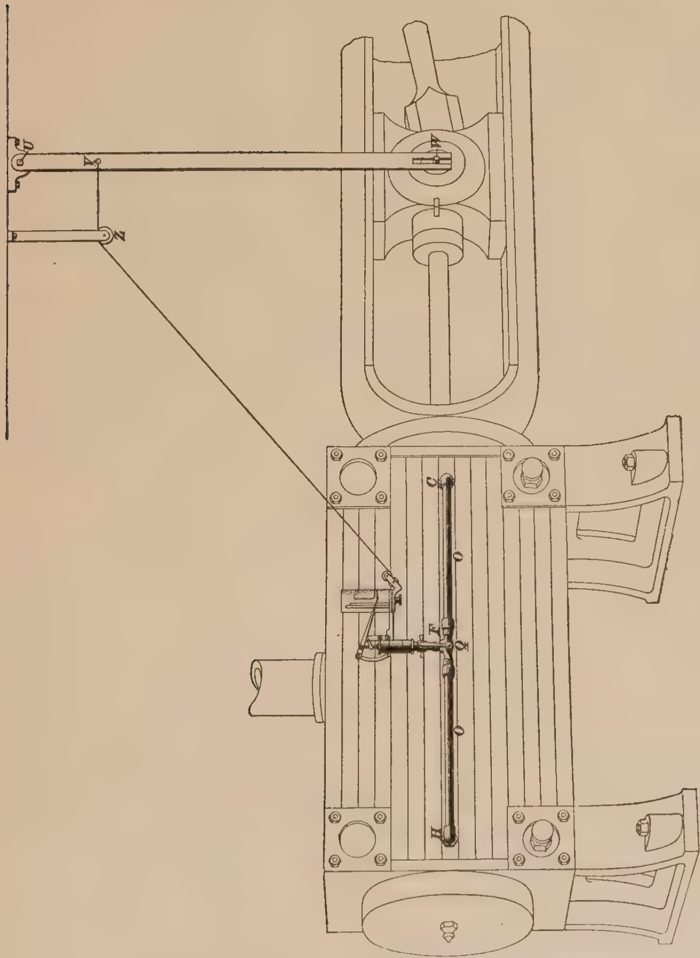


FIG. 6.

The lower end of the lever is slotted and fits over a pin in the crosshead; the other end of the lever is pivoted at a fixed point U and the cord is attached at V. The cord is

guided by a pulley Z so that it will leave the point V in a direction parallel to the line of motion of the crosshead.

In the device shown in Fig. 7, the lever is connected to the crosshead by a link $W D$. The cord is attached to the circumference of a grooved sector $E V F$, called a **Brumbo pulley.** The center of the circle of which the sector forms a part lies in the center line of the lever pivot U. The sector may be attached directly to the lever or it and the lever may be keyed to a short shaft in such a manner that the cord can be led directly to the indicator. Instead of the sector, the cord may be attached to a pin, as in the motion shown in Fig. 6; in this case a guide pulley similar to the pulley Z, Fig. 6, would be required.

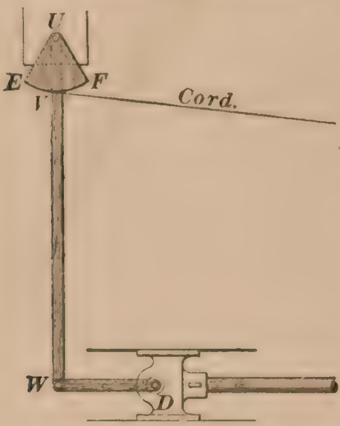


FIG. 7.

16. Error of Swinging-Lever Motions.—The types of swinging-lever motion illustrated in Figs. 6 and 7 are imperfect, from the fact that the motion imparted to the cord is not exactly proportional to the motion of the crosshead. In the slotted swinging lever, Fig. 6, the distance from the pivot U to the center of the pin in the crosshead is variable, while the distance from U to the point V to which the cord is attached is constant; in other words, the length of the long arm of the lever varies, while the length of the short arm remains constant. This results in a variation in the relative motions of crosshead and cord for different parts of the stroke, and the diagram obtained is, in consequence, distorted.

With the motion illustrated in Fig. 7, the link $W D$ acts like a connecting-rod to transmit the straight-line motion of the crosshead to the end of the lever that moves in a circular arc; the link thus has an angular motion that has a

disturbing effect on the ratio of the cord movement to that of the crosshead. The result is a diagram whose proportions are not perfect.

17. Methods of Reducing the Errors.—For ordinary work with the indicator, the amount of distortion with carefully made swinging-lever motions is not serious and may be ignored. To secure good results, the levers should always be suspended from such a point that when the crosshead is at the middle of its stroke, they will be perpendicular to its line of motion. The accuracy of the motion will, in general, be increased by increasing the length of the levers; for most purposes it will be sufficient to use a lever whose length is twice that of the stroke, and in some cases a lever even shorter than this is used. The accuracy of the motion with the connecting link, Fig. 7, is also increased by increasing the length of the link; for ordinary work, a link whose length is equal to one-third of the length of the stroke may be used. The lever in this motion should be so suspended that the extreme positions of *W* above and below the line of motion of the point *D* are about equal.

18. Theoretically Correct Motions.—The errors of the swinging-lever motions that were noted in Art. **16** can be neutralized and a theoretically correct reduction obtained by the methods illustrated in Figs. 8 and 9. In each of the figures the cord is attached to a sliding bar *S* whose line of motion is parallel to that of the crosshead. In Fig. 8 the bar *S* is provided with a pin that works in a slot in the swinging lever. By this arrangement the ratio of the distances from the pivot *U* to the pins *W* and *V* is constant for all positions of the crosshead, and the motion of the bar *S* is exactly

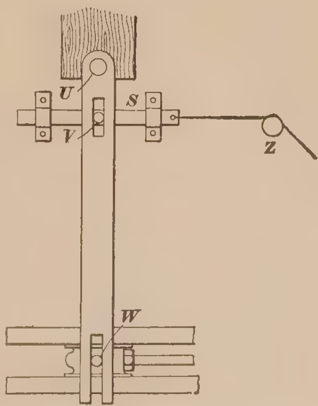


FIG. 8

proportional to that of the crosshead. In Fig. 9 the bar S is connected to the swinging lever by a short link $V\,C$. In order to secure a theoretically correct reduction, the length of this link must be such that the ratio of the distance $V\,C$ to the distance $W\,D$ is equal to the ratio of $U\,V$ to $U\,W$, and the bar S must be so located that the center lines $V\,C$ and $W\,D$ of the two links are parallel. When these conditions are fulfilled and the points U, V, and W lie in the same straight line, the center of the joint C lies on the straight line joining U and D.

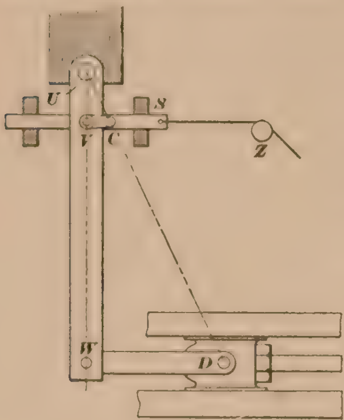


FIG. 9.

The cord must be guided by pulleys, as Z, so that it will leave the bars S, Figs. 8 and 9, in a direction parallel to the line of motion of the bars.

19. Rule for Proportioning Swinging-Lever Motions.—With any swinging-lever reducing motion, the ratio of the length of the diagram to the length of the stroke is equal to the ratio of the distance from the pivot to the point of attachment of the cord to the distance from the pivot to the pin by means of which the lever is connected to the crosshead; thus, with either of the motions shown in Figs. 6 to 9, let l represent the length of the diagram and L the length of the stroke, then $\frac{l}{L} = \frac{U\,V}{U\,W}$. In accordance with this principle, we have the following

Rule.—*To find the distance from the pivot at which to connect the cord, or to find the radius of the Brumbo pulley when the length of the stroke, the length of the diagram, and the distance from the pivot of the lever to the point where it is connected with the crosshead are known, multiply the length of the diagram by the distance from the pivot to*

the point of the lever at which it is connected with the crosshead and divide the product by the length of the stroke.

Let l = length of diagram;
L = length of stroke;
d = distance from pivot to point of attachment of cord (see $U\ V$, Figs. 6 to 9);
D = distance from pivot to point where lever is connected to crosshead (see $U\ W$, Figs. 6 to 9).

Then,
$$d = \frac{D\,l}{L}.$$

EXAMPLE 1.—The stroke of an engine is 28 inches; the length $U\ W$ of the lever is 6 feet; what must be the distance $U\ V$ to give a diagram $3\frac{1}{2}$ inches long?

SOLUTION.—Applying the rule just given, we have

$$d = \frac{72 \times 3\frac{1}{2}}{28} = 9 \text{ in. Ans.}$$

EXAMPLE 2.—In Fig. 7 find the radius $U\ V$ of the arc $E\ F$ in order that the diagram may be $3\frac{1}{2}$ inches long, the stroke of the engine being 38 inches and the length $U\ W$ being 5 feet 5 inches.

SOLUTION.—This example is solved in the same manner as the preceding one. The effective length of the lever is 5 feet 5 inches = 65 inches. Applying the rule, we have

$$d = \text{radius of arc } E\ F = \frac{65 \times 3.5}{38} = 6 \text{ in., nearly. Ans.}$$

REDUCING WHEELS.

20. Reducing wheels form a very convenient and theoretically accurate method of reducing the motion of the crosshead to the required value for the paper drum. These wheels are furnished in a variety of forms, some of which are designed to be attached directly to the indicator, while others are provided with means for clamping to some point on the engine bed. One of the latter type is illustrated in Fig. 10. The cord a is attached to a bar or rod fastened to the crosshead in such a manner that the cord will lead from it to the wheel in a line parallel to the line of motion

of the crosshead, and the cord *b* is attached to the indicator drum. The smaller pulley can be removed and replaced by one of several others of different sizes. The proportions of the two pulleys can thus be varied so as to secure the desired length of diagram. Thus, if the stroke of the engine is 12 inches and the desired length of the diagram is 3 inches, the diameter of the larger pulley should be four times that of the smaller. The hub of the larger wheel contains a spring that is wound up when the cord *a* is unwound from the wheel by the outward motion of the crosshead; when the crosshead makes its return stroke, the spring turns the wheel and winds the cord on again.

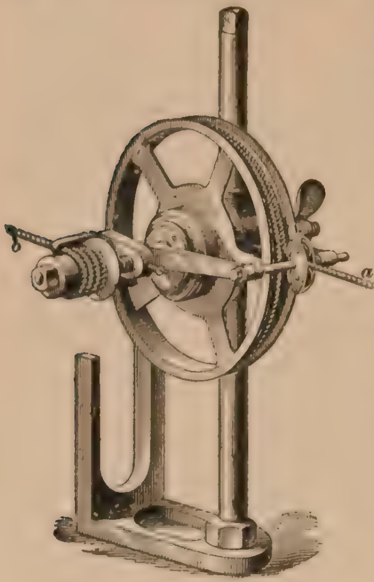


FIG. 10.

REDUCING MOTIONS FOR HIGH SPEEDS.

21. When an engine has a high rotative speed, the quick changes in direction of motion set up severe stresses in a reducing gear. In a pantograph or a swinging-lever motion, these stresses are likely to cause a springing of the parts that will distort the diagram and lead to erroneous results. The shocks and stresses also tend to wear the joints rapidly; the lazy-tongs motion, with its great number of joints and moving pieces, is on this account poorly adapted for high-speed work. A swinging lever, if made of stiff and light wood with joints bushed and neatly fitted, will give good results at nearly any speed of rotation at which it is practicable to run an engine. Instead of bushing the joints,

they may be made adjustable as shown in Fig. 11. The end of the bar is split by a saw kerf passing through the center of the hole forming the pivot bearing and extending far enough into the bar to permit the two parts to be drawn up tight against the pivot by a wood screw *S*. Instead of the screw a somewhat more substantial method is to use a small bolt passing through the end of the bar. A joint made in this way and fitted to a turned pin, if well lubricated, gives the best satisfaction and lasts almost indefinitely.

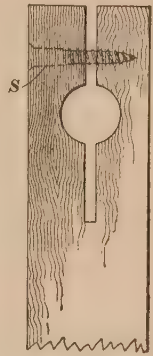


FIG. 11.

22. With reducing wheels the quick reversals of the direction of rotation that take place at high speeds make it necessary to use a stiff spring to overcome the inertia of the wheel. To reduce the inertia, the wheels are made as light as is practicable; with many reducing wheels, lightness is secured by the use of aluminum. The better class of reducing wheels can now be successfully used for nearly any speed of rotation likely to be met with.

INDICATOR CONNECTIONS.

23. Indicator Cords.—In order to transmit the motion from the reducing motion to the paper drum with as little loss or distortion as possible, it is necessary to use a cord that will stretch but little. To meet this requirement, indicators are generally supplied with a special braided cord that will give good results for most purposes. In the case of large engines, where long cords are required, the amount of stretch with the best cord obtainable is considerable and may result in a distortion that would be undesirable for accurate tests. For such cases a fine copper or steel wire may be used to advantage. It is always best to so arrange the reducing motion and indicator that the cord may be led to the paper drum in the shortest and most direct practicable

line. When the cord is attached to a pin on the reducing motion, it must be guided so as to leave the pin in the line of its motion, as is illustrated in Figs. 6, 8, and 9; the use of guide pulleys should, however, be avoided as much as it is practicable.

24. Stop Motions.—Various means are used to stop the motion of the paper drum when it is required to change the paper or when the indicator is not in use. A common method is to have a short cord attached to the paper drum with a hook *a*, Fig. 12, on the end; the cord from the reducing motion has a loop into which the hook may be fastened when it is desired to operate the paper drum. The length of the cord from the reducing motion can readily be made adjustable by the use of a loop *l*, formed as shown in Fig. 12. The thin strip of wood or metal *b* provides a very ready means of changing the length of the loop and of tying it securely in any position. To prevent the cord leading from the reducing motion from being thrown about and getting tangled when it is unhooked from the cord leading to the paper drum, it is well to have a rubber band fastened in a convenient position and provided with a hook into which the loop *l* may be secured. The elasticity of the rubber band can thus be made to keep the cord stretched and to prevent it from being tangled and broken.

FIG. 12.

25. Paper drums are sometimes provided with a stop motion that will hold them in place and prevent the cord from being wound on; this merely has the same effect as lengthening the cord, but is open to the objection that at high speeds the loose cord is apt to make trouble by flying about and getting caught. In addition to the stop motions above noted, indicator manufacturers have designed a number of very useful devices, some of which absolutely prevent any trouble with the cord and make it easy to start and stop the paper drum.

SPECIAL ATTACHMENTS.

26. Simultaneous Diagrams.—With an engine having two or more cylinders it is sometimes desirable to take a diagram simultaneously from each end of all the cylinders so as to get a record of what takes place in each cylinder at some particular time. It is difficult for a number of operators to apply the pencils of a set of indicators to the paper all at the same time; and, to overcome this difficulty, a number of devices have been invented by means of which the pencils of all the indicators can be simultaneously operated by one person. Of these devices the simplest and most successful is an electromagnet that is attached to the indicator. When a number of indicators are to be operated simultaneously, the electromagnets of all are connected by a wire; when a current is sent through the wire by pressing a button or key, each electromagnet pulls its pencil against the paper and holds it there until the circuit is opened.

INDICATOR DIAGRAMS.

DIRECTIONS FOR TAKING DIAGRAMS.

27. The makers of indicators furnish very complete instructions for the care and use of their instruments; these instructions should be carefully studied before attempting to use a new indicator or one with which the user is not thoroughly familiar. The following directions for taking diagrams apply to all makes of indicators: Before attaching the indicator to the engine, see that it is clean and in good working order. The piston should move freely. See that the joints of the various levers and links are oiled with fine oil and that they are slack enough to avoid friction, yet not so slack as to allow the pencil to shake. Adjust the pencil so that it just touches the paper and sharpen the point so that it makes a very fine light line. A heavy coarse line on a diagram indicates poor work.

Select a spring that will give a diagram about $1\frac{1}{2}$ or $1\frac{3}{4}$ inches in height. If, upon trial, the spring chosen makes a wavy line, choose a stiffer one. A stiffer spring is required on a fast-running engine than on a slow-running engine when the steam pressure is the same. See that there is no backlash between the piston and spring.

Adjust the length of the cord so that the drum turns backwards and forwards without striking either of the stops at the end of the travel. When it touches one or the other of the stops, the cord is either too short or too long. If it touches both, the travel of the drum is too great, and the cord must be fastened to a point on the reducing motion having less travel.

Keep the drum moving only when taking diagrams. Unhook the cord before putting a paper on the drum. In putting on the card, see that it fits the drum without wrinkles, and fold back the projecting edges over the clips *m*, Fig. 1, so that they will not touch the pencil lever.

28. Before taking the diagram, turn on the steam a minute or so to warm the indicator; then press the pencil lightly on the paper long enough to take a single diagram. Shut the cock and again press the pencil to the paper. Since the indicator piston is then only subjected to atmospheric pressure, the pencil will make a straight line called the atmospheric line. Disconnect the cord and remove the card. Write on the card the scale of the spring used, the speed of the engine, and any other desired particulars.

If one indicator is used for both ends, first open the three-way cock to admit steam from one end. Take the diagram and open the cock to the other end, and take the diagram from that end. Then shut off the steam entirely and take the atmospheric line.

GENERAL FEATURES.

29. Purpose.—In actual practice the imperfections in the construction of the engine and the velocity at which the steam must flow through the pipes and ports combine

to modify the pressures in the cylinder and, in consequence, the form of the diagram drawn by the indicator pencil. By a careful study of the peculiar features of the diagram, an experienced engineer is able to determine with a considerable degree of certainty the general type and condition of the engine and the circumstances under which the diagrams were taken. The following general outline of the characteristic features of diagrams taken under different conditions will enable the student to interpret most of the diagrams with which he will meet.

30. Points and Lines of the Diagrams.—In Figs. 13 and 14 are shown indicator diagrams from the crank end

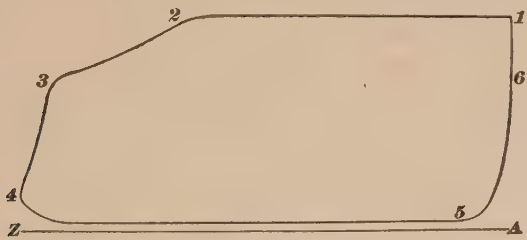


FIG. 13.

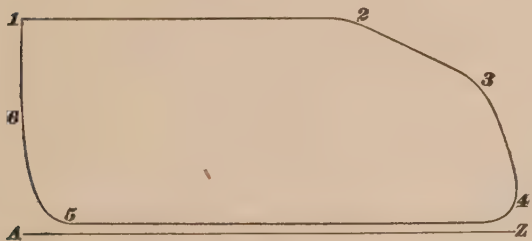


FIG. 14.

and head end of an engine. The different *points* of the stroke are plainly shown. They are as follows:

1 is the beginning of the stroke.
2 is the point of cut-off.
3 is the point of release.
4 is the end of the stroke.
5 is the point of compression.
6 is the point of admission.

The *lines* included between any two of these points have received special names, which are as follows:

6–1 is the admission line.
1–2 is the steam line.
2–3 is the expansion curve.
3–4–5 is the period of release.
4–5 is the back pressure line.
5–6 is the compression curve.
A Z is the atmospheric line.

31. The Vacuum Line.—It is sometimes desirable to have the vacuum line (line of no pressure) on the card also. The vacuum line may be drawn below and parallel to the atmospheric line. The distance between them will be $\frac{14.7}{\text{scale of spring}}$ inches. Thus, if the scale of the indicator spring is 30, the vacuum line lies $\frac{14.7}{30} = .49$ inch below the atmospheric line. When great accuracy is desired, the vacuum line should be located in accordance with the indication of the barometer. This is especially desirable when the engine is located at a great altitude above sea level. Then the distance between the atmospheric and vacuum lines may be found by multiplying the reading of the barometer in inches by .49 and dividing by the scale of the spring. For instance, if the barometer stands at 25 inches and the scale of the indicator spring is 30, the vacuum line should be drawn at a distance of $\frac{25 \times .49}{30} = .41$ inch from the atmospheric line.

32. Two Diagrams on a Single Card.—If but one indicator is used, the two diagrams are taken on the same blank, as shown in Fig. 15. With the diagrams placed one over the other, as shown, it is very easy to tell what is taking place in the cylinder at any point of the stroke. On the forward stroke the pencil of the indicator describes the line *A B C D* of the head diagram if the cock is opened to

the head end, or it describes the line KLM if the cock is opened to the crank end. Likewise, the lines $GHJK$ and DEF are described during the return stroke.

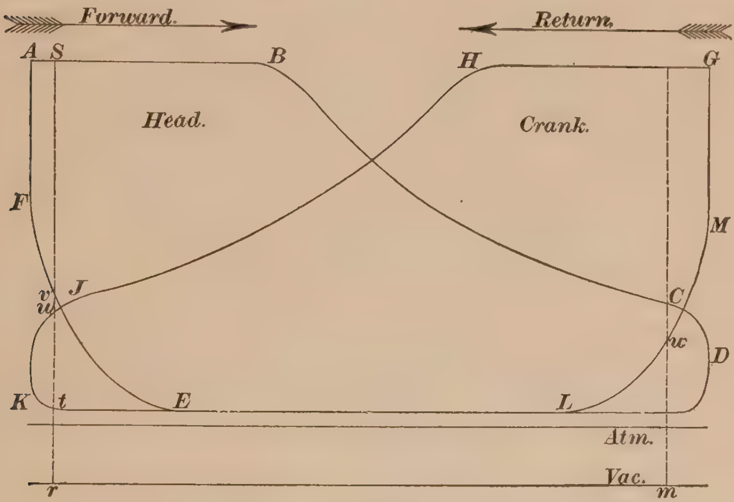


FIG. 15.

Suppose the piston is at a position corresponding to r on the forward stroke; the pressure (above vacuum) urging the piston forwards is $r\,S$, while the pressure resisting is $r\,t$. Hence, the net pressure on the piston is $S\,t$. Suppose, now, that the piston is at r on the return stroke; the pressure at the right urging the piston on is $r\,u$, while the pressure on the left is $r\,v$. The net pressure is, therefore, $u\,v$, and is negative; or, in other words, the resistance is greater than the effort.

33. A double diagram of this character tells at a glance what is taking place at either end of the cylinder at any point of the stroke. Thus, when the piston is on the forward stroke, in the position corresponding to m, the steam in the head end is at the point of release C. Draw a line through m perpendicular to the vacuum line. C lies on $A\,B\,C$, and since $K\,L\,M$ is described at the same time as $A\,B\,C$, the intersection of the line through C with the

line $K L M$ is the point corresponding to C. The intersection w is on the compression line; hence, when release occurs in the head end, compression is taking place in the crank end.

34. Effect of Type and Speed of Engine on Form of Diagram.—The form of a good diagram depends largely

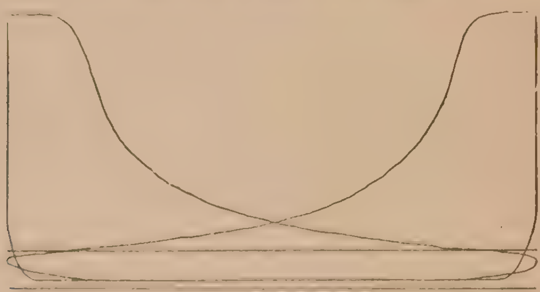


FIG. 16.

on the type of the engine, style of valve, and speed. What would be considered a good diagram from a locomotive or from a high-speed automatic engine would be considered very poor if taken from a Corliss engine. In general, a diagram taken from an engine with releasing gear of the Corliss type will be regular and show but little compression.

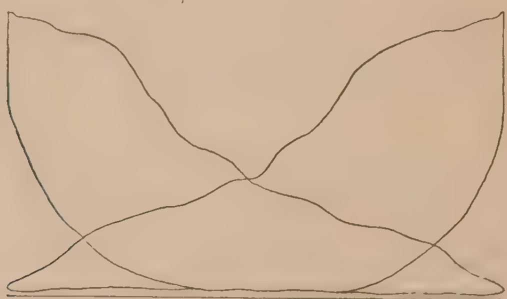


FIG. 17.

The point of cut-off, release, and compression will be sharply marked. The diagram shown in Fig. 16 is what may be

expected from this type of engine when the valves are correctly set and in good working order. The fact that the back-pressure line runs below the atmospheric line shows plainly that the engine the card was taken from was condensing. On the other hand, Fig. 17 shows the form of diagram that may be expected from an engine running at 250 to 300 revolutions per minute. On account of the high rotative speed, the lines are irregular, due to the inertia of the moving parts of the indicator. The compression is large, as it should be for engines running at a high speed. The point of cut-off is never very sharply marked.

It is readily seen how totally unlike are the diagrams shown in Figs. 16 and 17, yet each is considered as representing good practice.

FAULTS IN VALVE SETTING REVEALED BY DIAGRAMS.

FAULTS IN STEAM DISTRIBUTION.

35. Some of the most common faults revealed by the indicator diagram are given below. In the diagram following, Figs. 18 to 22,

1 is the admission.
2 is the cut-off.
3 is the release.
4 is the compression.

I. Admission may be too early.
II. Admission may be too late.
III. Cut-off may be too early.
IV. Cut-off may be too late.
V. Release may be too early.
VI. Release may be too late.
VII. Compression may be too early.
VIII. Compression may be too late.

36. Case I. Admission Too Early.—The effect on the diagram of a too early admission is shown in Fig. 18. It is

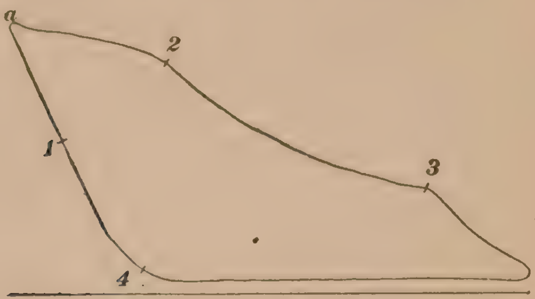


FIG. 18.

seen that the admission line *1 a*, instead of being straight and perpendicular to the atmospheric line, as in Figs. 13 to 16, slants backwards. With a single slide valve or piston valve, all the other events, cut-off, release, and compression, are also too early. The remedy is to shift the eccentric on the shaft so as to decrease its angular advance.

In the case of the Corliss engine, the admission may be too early, while the other points are not affected. The fault may then be remedied by adjusting the link rods so as to give the steam valves more lap, and it may not be necessary to shift the eccentric.

The effect of too early admission is to introduce an excessive resistance to the motion of the piston before it reaches the end of the stroke; in consequence, the piston must be pushed to the end of its stroke by the momentum of the flywheel acting through the crank and connecting-rod. The result is likely to be pounding at the crosshead, crank, and main bearing.

37. Case II. Admission Too Late.—In this case the admission line *1 a* on the diagram slants forwards, as shown in Fig. 19. The remedy is to increase the angular advance until the admission line *1 a* becomes perpendicular to the atmospheric line. With a single slide valve in the case of a

too late admission, the other events, as *2*, and particularly *3* and *4*, are also too late.

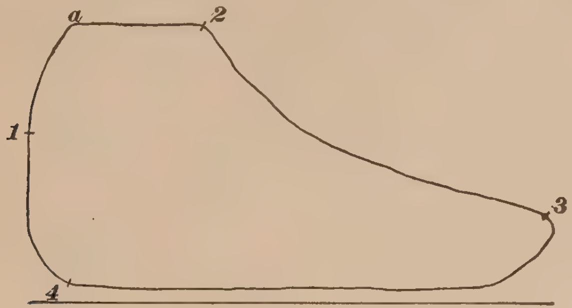


FIG 19.

In case the engine has a Corliss or other releasing gear, the admission may be made earlier by reducing the lap of the admission valves.

The effect of late admission on the running of the engine is not generally as noticeable or severe as is too early admission. It is, however, generally desirable to give the valves enough lead to have the clearance space filled with steam at the boiler pressure just as the piston begins its stroke.

38. Case III. Cut-Off Too Early.—See Fig. 20. Here the steam expands until its pressure is less than the back

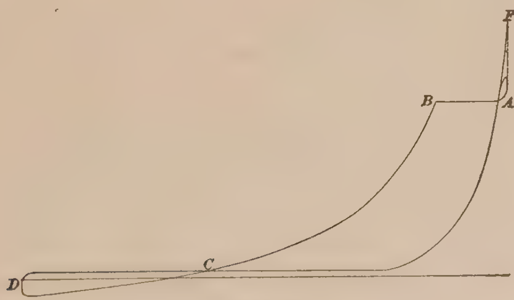


FIG. 20.

pressure; in consequence, the expansion line crosses the back-pressure line, as shown at *C*, and forms a loop. This effect is often observed in automatic cut-off engines working

under a light load. It causes a reversal of the pressures on the piston that may result in pounding. The great range in pressure also has a bad effect on the steam consumption.

39. Case IV. Cut-Off Too Late.—See Fig. 21. Here it will be noticed that the terminal pressure is very high.

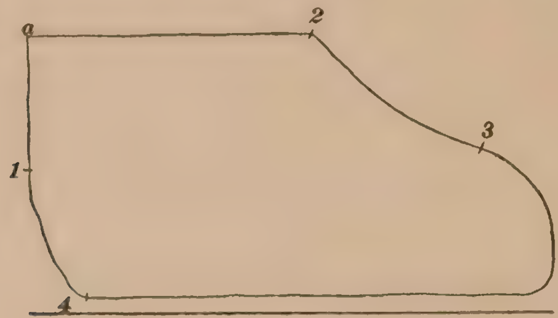


FIG. 21.

When this is the case, a great deal of the benefit of expansion is lost and there is a consequent waste of steam. With an ordinary plain slide-valve engine, the cut-off is always late, it not being practicable to cut off earlier than ½ stroke without seriously affecting the other events.

40. Case V. Release Too Early.—The appearance of the diagram for this case is illustrated in Fig. 18.

41. Case VI. Release Too Late.—This is illustrated in Fig. 19.

42. Case VII. Compression Too Early. — Figs. 20 and 22 show the effects of too early compression. The steam is compressed in the clearance space until its pressure rises above that of the steam in the steam chest; when the steam valve opens there is a flow of steam from the cylinder to the steam chest, as is shown by the loop, until the pressures in the cylinder and steam chest are nearly equal. If the steam valve has no lead, the compression line may rise above the admission line, as shown at *F* in Fig. 20; with lead, the loop will have the form shown in Fig. 22 and at *A* in Fig. 20.

Too much compression is likely to produce an effect on the running of the engine similar to that produced by too early admission; pounding and heating are often to be ascribed to this cause. It also reduces the effective work of the steam.

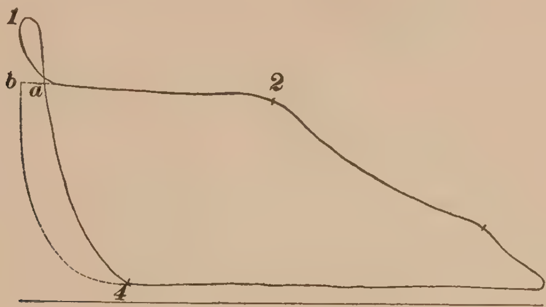


FIG. 22.

With the same cut-off and the proper amount of compression, an amount of work that is represented by the area *a b 4 a*, Fig. 22, included between the line *4 a* and the dotted line *a b 4*, plus the area of the loop, would be gained.

43. Case VIII. Compression Too Late. — This is shown on the diagram by a very short compression curve (see *4–1*, Fig. 19) or by the almost complete absence of such a curve.

REMEDIES FOR FAULTS IN STEAM DISTRIBUTION.

44. Most of the faults enumerated in the preceding articles are due either to incorrect valve proportions or to a fault in the setting; the remedy to be applied in any particular case will, therefore, be determined by a careful consideration of the type of engine and the conditions under which the diagram was taken. With a plain slide valve driven by a fixed eccentric, a change in the angle of advance of the eccentric will have an effect on all the events. Increasing the angle of advance will make admission, cut-off, and release take place earlier and increase the compression; decreasing the angle will have the opposite effect. A general rule is that an increase in the angle of

advance of any eccentric with a fixed throw has the effect of making all the events controlled by that eccentric take place earlier, while a decrease in the angle will make them take place later. This rule applies to all types of valves and gears.

It is generally desirable to set the valves so that the work done will be equally divided between the two ends of the cylinder. If an indicator diagram shows a material difference in the work done in the two ends of a cylinder with a slide valve, the fault can be remedied by changing the length of the valve stem so as to make the cut-off take place earlier on the end doing the greater amount of work and later on the other. With an engine having a separate steam valve for each end of the cylinder, either valve may be adjusted so as to make it cut off earlier or later without affecting the other.

45. Remedy for Too Early Cut-Off.—The fault illustrated in Case III and Fig. 20 cannot be remedied by a change in the valve. It is found only in the case of automatic or adjustable cut-off engines working with a light load and a high boiler pressure. The cause is a steam pressure too great for the work to be done with the given size of cylinder and piston speed. The remedy is either to throttle the steam, lower the boiler pressure, or run the engine at a slower speed. That any of these remedies will have the desired effect will be made clear by a consideration of Fig. 23.

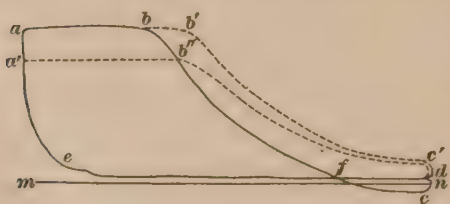


FIG. 23.

The solid line *a b c* represents the expansion line of a diagram when cut-off is so early that the steam expands below the back-pressure line *d e*. The work done is represented by the area *a b f e* minus the area *f d c*. Now let the steam be throttled or the boiler pressure be lowered so that the initial pressure will rise only to *a'*. In order that the work done in the cylinder may be the same with this

pressure as it was with the higher pressure, it is evident that cut-off must take place enough later to raise the expansion line $b'' c'$ far enough above the expansion line $b\ c$ to give the diagram $a'\ b''\ c'\ d\ e\ a'$ an area equal to the net area represented by the full-line diagram. On account of the later cut-off, the terminal pressure does not fall below the back-pressure line and no loop is formed.

If the number of revolutions per minute is reduced, the total work done by the engine remaining constant, it is evident that the work done during each stroke must be increased; this will require the admission of steam during a greater portion of the stroke, so as to produce a diagram having a greater area; expansion will begin later and the expansion line will be prevented from falling below the back-pressure line, as is indicated by the dotted line $b'\ c'$.

46. Remedy for Too Late Cut-Off.—With an *adjustable cut-off*, the fault illustrated by Fig. 21 can be remedied either by raising the boiler pressure so that the same area of diagram will be obtained with an earlier cut-off or by increasing the number of revolutions per minute so as to do the same work with a smaller average pressure. By either method the cut-off will be made to take place earlier in the stroke, and the expansion line will, in consequence, fall nearer to the back-pressure line.

47. Release Too Early or Too Late.—If the release is too early, there is danger of loss of pressure due to the escape of the steam too early in the stroke; on the other hand, if the release is too late, the escape of the steam will be so much retarded that the back pressure at the beginning of the return stroke will be excessive. Either of these will represent a loss of work. The valve should be so designed and set that the drop from the expansion line to the back-pressure line will occur as nearly as possible at the end of the stroke. If the engine is provided with separate steam and exhaust valves, this condition will best be fulfilled by setting the exhaust valves so that one-half of the fall in pressure occurs before the piston begins its return stroke;

the release line will then have a form that is well shown in Fig. 15. With a single-valve engine it is often very difficult to secure a satisfactory release line without seriously affecting the other events controlled by the valve.

48. Rules for Compression.—The best indication that the amount of compression is satisfactory is a quiet- and cool-running engine. At the end of the stroke the reciprocating parts must have their direction of motion changed; a force must act to stop them and reverse their direction of motion. By closing the exhaust before the end of the stroke, a part of the steam that would otherwise escape from the exhaust pipe and be lost is retained in the cylinder and acts as a cushion that helps to stop the reciprocating parts quietly. The energy given up by the reciprocating parts in being brought to rest, instead of being wasted in the production of knocks that would result in heating and rapid wear in the bearings, is stored in the steam compressed in the clearance space. The clearance space is thus filled with steam at a pressure more nearly equal to the boiler pressure, and the quantity of steam that must be taken from the boiler is correspondingly reduced.

If there is too little compression, the reciprocating parts will not be satisfactorily cushioned; if there is too much compression, the energy due to the motion will be absorbed before the end of the stroke; the piston must then be pushed by the crank. In either case the effect will be a sudden reversal in the pressures on the bearings that will produce shocks and heating.

49. No simple rule for determining the exact amount of compression to use can be given; however, it may be stated that the amount of compression required to secure quiet running varies with the speed of the engine, but in no case should the compression line extend above the initial or boiler pressure.

It is average practice to compress to about $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{5}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines.

OTHER FAULTS REVEALED BY DIAGRAMS.

50. Introduction.—In addition to faults in valve design and setting, indicator diagrams furnish useful indications of the condition of the piston, cylinder, and valves; insufficient port opening, a steam pipe too small for its purpose, a cramped exhaust passage, or an obstructed exhaust pipe also have an effect on the appearance of the diagram that will generally make it comparatively easy to locate the fault. For a location of some of these faults, it is necessary to draw in the theoretical expansion line, or equilateral hyperbola.

51. Constructing the Theoretical Expansion Line. The general method of using the equilateral hyperbola for testing the expansion line of an actual indicator diagram is illustrated in Fig. 24, where the diagram $E\ C\ K\ L$ and the

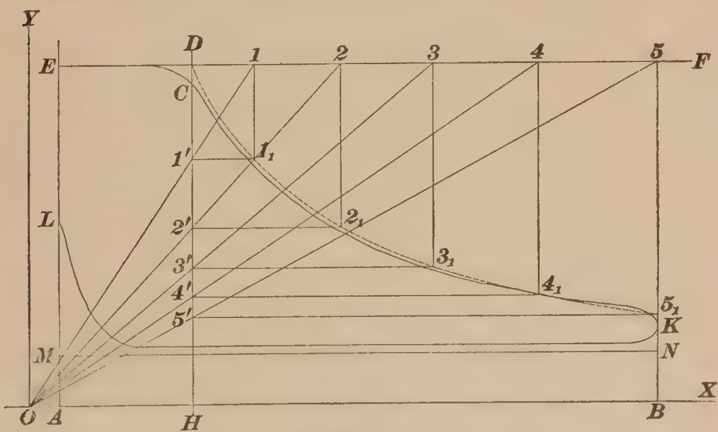


FIG. 24.

atmospheric line $M\ N$ represent the lines drawn by the indicator. To draw the theoretical expansion line on this diagram, first draw the vacuum line $O\ X$, as explained in Art. **31.** Perpendicular to $M\ N$ and $O\ X$ draw the two lines $A\ L$ and $B\ K$, just touching the two ends of the diagram. Measure the length $A\ B$ between these two perpendiculars, and this will give the length of the diagram. Multiply the length so found by the clearance volume of the

end of the cylinder from which the diagram was taken, expressed in per cent., and divide by 100; lay off from *A* a distance *A O* equal to the quotient. From *O* draw the perpendicular *O Y*; this is the **clearance line.** Through the highest point *E* of the steam line draw the horizontal line *E F*. Locate, as nearly as may be done by inspection, the point of cut-off *C*, and through this point draw the perpendicular *D H*. The point *O* where the vacuum line *O X* (the line of no pressure) and the clearance line *O Y* (the line of no volume) intersect represents the point of no pressure and no volume; the distance *A E* or *H D* represents the original absolute pressure; and *O H* represents the original volume of the steam. To obtain points on the theoretical expansion curve, draw lines as *O 1*, *O 2*, *O 3*, *O 4*, *O 5* at random from *O* to the line *E F*. From the points of intersection of these random lines with the line *D H*, as the points *1′*, *2′*, *3′*, *4′*, and *5′*, draw lines parallel to the atmospheric line *M N*. Then, from the points of intersection *1*, *2*, *3*, *4*, and *5* of the random lines drawn from *O* with the line *E F*, drop perpendiculars intersecting the lines drawn from *1′*, *2′*, *3′*, *4′*, and *5′* in 1_1, 2_1, 3_1, 4_1, and 5_1. These intersections are points on the theoretical expansion line; consequently, through them, by means of an irregular curve, trace the line $D–1_1–2_1–3_1–4_1–5_1$.

52. Relation Between the Theoretical and the Actual Expansion Lines.—Numerous tests have shown that when the piston and valves are tight so as to prevent leakage of steam to or from the cylinder after cut-off takes place, the actual expansion line will agree very closely with the theoretical. It will generally be found that the actual line falls somewhat lower than the theoretical, especially towards the end of the stroke, and then raises above the theoretical expansion line, as is shown in Fig. 24 by the crossing of the two lines near 4_1. This is thought to be due to the phenomena known as **cylinder condensation** and **reevaporation,** which may be explained as follows: During the period of exhaust, the cylinder walls are cooled by contact with the relatively cool low-pressure exhaust steam. When

the hot steam from the boiler is admitted to the cylinder, a part of it condenses and gives up its latent heat to the cold walls and thus heats them again nearly to the temperature corresponding to the initial pressure; the water formed by this process of condensation is deposited in a thin film on the walls of the cylinder. After cut-off takes place and expansion begins, the pressure of the steam in the cylinder falls until its corresponding temperature is lower than the temperature of the cylinder walls; the walls then give up some of their heat and reevaporate some of the water. The steam thus formed towards the end of the stroke prevents the pressure from falling as fast as it otherwise would and has the effect of raising the actual expansion line somewhat above the theoretical.

53. Leaks Indicated by Expansion Line.—If the actual expansion line departs considerably from the theoretical, it is to be inferred that steam either enters or leaves the cylinder after cut-off takes place. Thus, in Fig. 25,

FIG. 25.

where the expansion line rises very markedly above the theoretical curve, it is evident that the valve leaks and allows steam to enter after cut-off. Similarly, if the expansion line fell below the theoretical curve, the inference would be that steam was escaping from the cylinder through a leaky exhaust valve or past an imperfectly fitting piston. An expansion line that closely follows the theoretical curve is not, however, conclusive evidence that the valves and piston are tight; steam may leak into and out of

the cylinder at the same time and at such rates that the expansion line will appear to be quite satisfactory.

54. Determining the Point of Cut-Off.—It is sometimes very difficult to determine exactly the point of cut-off from the indicator diagram, especially when the engine has a high rotative speed. The most general method of determining it is to prolong the expansion line upwards by means of an irregular curve and note where it leaves the actual line of the diagram; then take the point of cut-off at or very near the point of deviation (see Figs. 24 and 25).

Instead of locating the point *D*, Fig. 24, at which to begin the theoretical curve by the method just explained, a method sometimes recommended is to prolong the expansion line by means of an irregular curve until it intersects the horizontal through the point representing the initial pressure. The point of intersection is then taken as the point through which to draw the vertical line *D H*, Fig. 24, to represent the volume at point of cut-off.

The rounding of the diagram near the point of cut-off is caused by the slowness of the valve movement at cut-off. On the Corliss and other releasing-gear engines, the valve cuts off very suddenly, the rounding is very slight, and the point of cut-off is very easily located.

55. Leaks Indicated by Compression Line.—If the piston and valves are tight, the compression line will generally curve quite regularly in one direction until it meets the admission line, as is shown by the diagrams in Figs. 15, 16, and 17. It often happens that, as shown at *5* in Fig. 26, the curvature of the compression line changes as the piston nears the end of the stroke; this change sometimes becomes so pronounced as to form a

Scale of Spring=50.

FIG. 26.

hook shown at *a*, Fig. 27. A compression line of this form indicates that steam is escaping from the compression

FIG. 27.

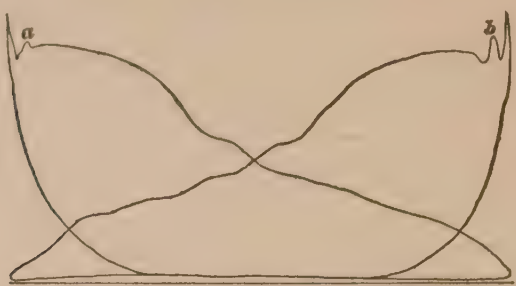
space; the loss may generally be ascribed to a leak either through the exhaust valve or around the piston.

56. Effect of Throttling.—When the steam pipe and its connections are of ample size and the steam ports are well opened, a nearly horizontal steam line whose height above the atmospheric line represents a pressure nearly equal to the boiler pressure may be expected, as is shown in Figs. 13 to 16. Any restriction in the passage leading from the boiler to the cylinder has the effect of preventing the flow of steam fast enough to keep the cylinder filled at boiler pressure up to the point of cut-off. This effect is shown on the diagram by a steam line that gradually falls as the piston advances. Fig. 26 is a diagram from an engine with a throttling governor; the effect of the governor in checking the flow of steam to the cylinder is shown by the drop in the steam line between the point *1* and the point *2*, where cut-off takes place. A long steam pipe or a pipe that is too small for its purpose, a partly closed throttle valve, or any similar obstruction will produce a drop in the steam line similar to the one shown in Fig. 26.

A high rotative speed generally results in a drop in the steam line, as is shown by the diagrams in Fig. 17. With shaft-governor engines, especially, the valve opening is often restricted and steam cannot follow the piston fast enough to keep the pressure up to that at the beginning of the stroke.

57. A high back pressure is caused by some obstruction that prevents the free escape of the exhaust. The exhaust from the engine from which the diagram shown in Fig. 26 was taken was discharged into a system of pipes for heating the building, and considerable pressure, as is shown by the height of the back-pressure line *3–4* above the atmospheric line *m n*, is required to force the steam through the coils of pipe. Somewhat similar results will be produced by a choked exhaust port or an exhaust pipe that is very long or too small.

58. Wavy lines on a diagram are generally due to vibrations of the pencil motion when there is a sudden change in pressure, such as takes place when steam is admitted to the cylinder of a high-speed engine or during the period of expansion. The indicator piston and the pencil motion are quickly set in motion and their inertia carries them beyond the point they would reach if the pressure were



50 Spring, 210 R.P.M., 12"x18" Engine.

FIG. 28.

gradually applied. This effect is well illustrated by the wavy lines at *a* and *b*, Fig. 28, which were produced by vibrations set up by the action of the steam at admission. The expansion lines of the same diagrams show a similar, but less violent, vibration. These effects are common with diagrams from quick rotative speed engines. They are an indication that the indicator piston is in good condition and working freely.

59. Sticking of the indicator piston is suggested by an expansion line that drops by a series of steps somewhat resembling notches (see Fig. 29). These steps or notches

FIG. 29.

can generally be distinguished from the wavy lines produced by the inertia of the parts of an indicator with a free working piston by their angular appearance.

60. Striking of the paper drum against the stops is readily detected by the appearance of the release end of the diagram, which, instead of being rounded, as shown in Fig. 27, will have angular corners with a nearly vertical line connecting them, as shown in Fig. 29, where the full line *a b* shows the appearance of the end of the diagram when the drum struck the stop; the dotted line shows the change effected by giving the drum its full range of motion.

ENGINE TESTING.

THERMODYNAMICS OF THE STEAM ENGINE.

INTRODUCTION.

1. Thermodynamics is that branch of physical science that treats of the relation between heat and mechanical work.

2. A **heat engine** is a device by means of which energy, in the form of heat, developed by the combustion of fuel or derived from any other source is transformed into mechanical motion in such a manner that it can be made to do useful work.

3. The Steam Engine a Heat Engine.—The steam engine is in reality a heat engine, steam being merely a vehicle by means of which the heat energy developed by the combustion of the fuel in the furnace is transferred to the moving parts of the engine. From these moving parts the energy is transmitted by such vehicles as shafting, pulleys, belts, and electric currents to the point where it can be made do the required work.

The general principle on which the action of nearly all heat engines is based is the production of an expansive gas or vapor in a confined space. With the steam engine, water is heated in a closed boiler and changed to an expansive

vapor—steam—whose pressure depends on the temperature to which it is heated. Other easily vaporized liquids, for example, naphtha, are sometimes used instead of water, and the action of their vapors depends on exactly the same principles as govern the action of steam.

4. Work Done by Expansive Force of a Gas.—All gases possess the property of expansibility, by virtue of which they expand and fill the space in which they are confined, no matter how great that space may be. This tendency to expand causes the gas to exert a pressure, called the **tension** of the gas, on the walls of the confining vessel. Keeping this principle in mind, let us consider a given volume of gas confined in a cylinder fitted with a movable piston. The gas in the cylinder tends to expand and thus exerts a pressure on the piston. If the force that resists motion is less than this pressure, the piston will be pushed outwards against the opposing force; the expansive force of the gas overcomes a resisting force, and in so doing does work.

5. Heat the Source of Work Done During Expansion. Careful experiments have shown that there is a fixed relation between work and heat and that heat can be changed into work and work into heat. A study of the effect on the gas of its expansion in the cylinder under such conditions that it does work will show that the work has really been done by heat.

To show that heat is the force that moves the piston, let the cylinder be so made that no heat can get to the gas as it expands; under these conditions a thermometer in the gas would show that it gets colder as it expands and pushes the piston along. The work has been done at the expense of a part of the heat of the gas and its temperature falls. In accordance with the theory of heat, the fall in temperature means that the molecules of the gas move slower; part of the kinetic energy represented by their rate of motion at the beginning of expansion has been expended in doing the work of pushing the piston against the resisting force.

6. If the cylinder is so arranged that enough heat can be added to the gas during expansion to keep its temperature *constant*, and a careful measurement of the heat added and the work done is made, it is found that the quantity of heat added is exactly equal to the heat represented by the work. For example, if the piston is pushed 4 feet by the expanding gas against an average resistance of 5,000 pounds, the work done is $4 \times 5,000 = 20,000$ foot-pounds. Since 778 foot-pounds of work is equivalent to 1 B. T. U. of heat, it follows that $20,000 \div 778$, or 25.707 B. T. U. of heat must be added to the gas to make up for the heat expended in pushing the piston and to keep the temperature constant.

7. Adiabatic Expansion.—When a gas expands and does work at the expense of its own heat—no heat being added to it from an outside source—the *expansion* is said to be **adiabatic.**

8. Isothermal Expansion.—When heat is added to a gas so as to keep the temperature constant during expansion, the *expansion* is said to be **isothermal.**

9. Compression of a Gas.—If we have a quantity of gas in a cylinder and push the piston inwards, so as to compress the gas and make it occupy a smaller space, we must do work in overcoming the expansive force of the gas; this work represents a certain amount of energy that is transferred to the gas. If the compression takes place under such conditions that no heat can leave the gas during the change in its volume, the energy represented by the work done on it will appear as heat and the temperature of the gas will be raised; under these conditions we have **adiabatic compression.**

If the compression takes place under such conditions that the heat represented by the work done is removed from the gas so as to keep its temperature constant, the *compression* is **isothermal.**

10. Relation Between Expansion and Compression. The quantity of work that must be done in compressing

a gas adiabatically or isothermally from a given volume to a smaller one is exactly equal to the work that the gas can do when expanding, in the same way in which it was compressed, from the smaller volume to the original. Also, the rise in temperature during adiabatic compression and the quantity of heat that must be abstracted when the compression is isothermal are, respectively, equal to the corresponding fall of temperature and the quantity of heat that must be added during adiabatic and isothermal expansion.

11. Relation Between Work and Heat During Expansion or Compression.—In practice, it is seldom that the expansion is purely adiabatic or isothermal. No cylinder can be so made as to absolutely prevent the transfer of some heat to or from the gas, and it is difficult to impart or abstract heat so as to keep the temperature uniform. In any case, however, it is always found that there is a definite relation between the work done and the sum of the quantities of heat represented by the change in temperature of the gas and the heat imparted to or abstracted from it. This relation shows conclusively that the work done by an expanding gas is always a change of heat to work.

12. Expansion Diagrams.—The relation between the pressure and the volume of a gas during expansion may be represented by means of a graphical diagram. To illustrate, consider a cylinder A, Fig. 1, in which a piston P fits. The cylinder is attached to a reservoir R by a pipe T that permits air from R to enter the space S when the valve V is opened. A gauge G graduated so as to indicate absolute pressures, that is, so that the pointer stands at zero when there is a perfect vacuum in the space S, shows the pressure in the cylinder; a cock C, when opened, permits any air in the cylinder to escape when the piston is pushed back. Now, with the valve C open, push the piston clear back to the end of the cylinder, thus forcing out all the air; then close C and open V, so as to admit air from R, in which there is a constant pressure. Permit the piston to move slowly to the left with V open, and the gauge shows a constant pressure

of the air in the space S. When the piston has moved a certain distance to the left, close V, so as to stop the admission

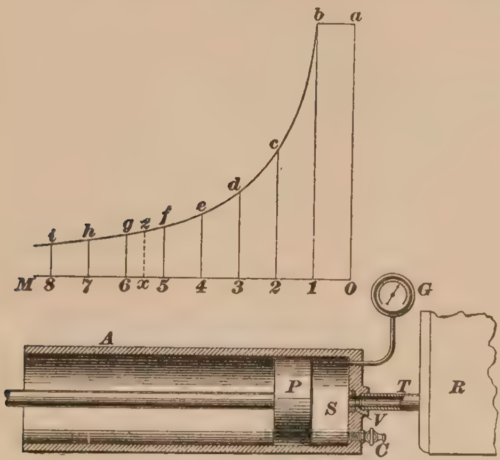


FIG. 1.

of air from the reservoir. Now, as the piston is permitted to move farther to the left, the gauge shows that the pressure falls. If the temperature in the cylinder is kept constant, it is found that when the piston is twice its original distance from the end and the air has expanded to twice its original volume, the pressure, in accordance with Mariotte's law, is only one-half the original pressure. When the volume is three times as great as the original volume, the pressure is found to be one-third the original pressure. When the volume has increased four times, the pressure is one-fourth as great, etc.

13. To represent this action graphically, draw a line $O\,M$ to represent the piston motion and divide this line into a number of equal parts, O–1, 1–2, 2–3, etc., each of which, to some convenient scale, represents a motion of the piston through a distance equal to that through which it moved while V was open. Since the volume of air in the cylinder is proportional to the distance of the piston from the end of the cylinder, each of the sections O–1, O–2, etc. represents

a volume equal to the original volume of air admitted to the cylinder from the reservoir, and the distances *O–1*, *O–2*, *O–3*, etc. represent the volume of the air in the cylinder for piston positions corresponding to the points *1*, *2*, *3*, etc.

From *O* draw a vertical line *O a*, and, to some convenient scale, make its length represent the pressure at the beginning of the piston stroke. Draw other vertical lines from the points *1*, *2*, *3*, etc., and, to the same scale as that to which *O a* was drawn, make their lengths represent the pressures corresponding to the piston positions represented by the points *1*, *2*, *3*, etc. and to the volumes represented by the distances *O–1*, *O–2*, *O–3*, etc. Since the pressure, when the piston is at *1*, is the same as the pressure at the beginning of the stroke, the length of the perpendicular *1–b* is the same as the length of *O a*. At *2* the volume is *O–2*, twice the original volume, and if the expansion is isothermal, the pressure is one-half the pressure at *1*; consequently, the length of the line *2–c* is one-half the length of *O a* or *1–b*.

Any desired number of points *c*, *d*, *e*, *f*, etc. can be located and a curve drawn through them. The distance of any point *x*, on the line *O M*, from the point *O* represents, to the scale of volumes, the volume of air in the cylinder when the piston is in the position corresponding to this point; likewise, the vertical distance *x z* from the point *x* to the curve represents, to the scale to which the pressures were laid off, the pressure for the corresponding piston position and volume.

14. The Isothermal Expansion Line, or Equilateral Hyperbola.—The curve that represents the relation between the pressure and volume when the temperature is constant is called the **isothermal expansion line.** This curve follows the law of the curve known in mathematics as the **equilateral hyperbola**; it is, therefore, often called by that name.

15. The Adiabatic Expansion Line.—If no heat is added to the air as it expands, that is, if the expansion is adiabatic, the gauge *G*, Fig. 1, shows a more rapid drop in

pressure as the piston advances; each vertical line representing the pressure in the cylinder after expansion begins is shorter than the corresponding line of the isothermal curve; the curve drawn through their upper ends will, therefore, fall below the isothermal curve. The curve representing adiabatic expansion is called the **adiabatic expansion line.**

16. Comparison of the Isothermal and Adiabatic Expansion and Compression Lines.—In Fig. 2, the

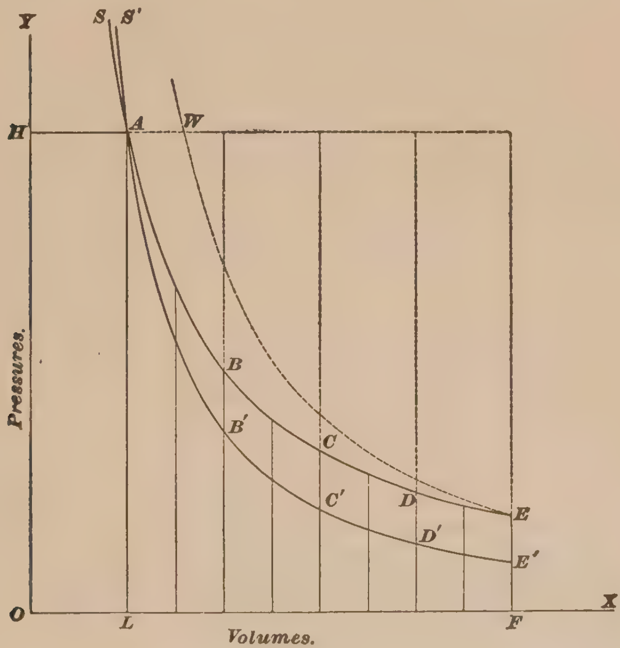


FIG. 2.

curve $A\ B\ C\ D\ E$ represents the isothermal, and $A\ B'\ C'\ D'\ E'$ the adiabatic, expansion line of a quantity of air whose original volume and pressure are, respectively, represented by the distances $O\ L$ and $O\ H$ or $L\ A$. If the air were compressed isothermally from the volume $O\ F$ and the pressure $F\ E$, the pressure would rise as the volume decreased, and the curve $E\ D\ C\ B\ A$ would show the relation between the

volume and the pressure. If, with the same original volume and pressure, the air were compressed adiabatically, the curve representing the relation between the volume and the pressure would rise above the isothermal compression curve, as is shown by the dotted line $E\,W$. If a quantity of air, whose volume is represented by $O\,F$ and whose pressure by $F\,E'$, is compressed adiabatically, the curve representing the relation between the volumes and the pressures during the process of compression will be $E'\,D'\,C'\,B'\,A$, which is the same curve that represented the relation for adiabatic expansion from the volume $O\,L$ and the pressure $L\,A$.

17. Expansion of Steam.—When steam expands and does work, there is the same relation between heat given up and work done as has been explained for gas. Owing, however, to the properties of saturated steam, by virtue of which the pressure depends solely on the temperature and is independent of the volume, the relation between volume and pressure is not as simple as is the case with a perfect gas. For example, if a given weight of dry saturated steam expands adiabatically, a part of it will be condensed; while if the expansion is isothermal, the steam will be superheated during its expansion. If there is a mixture of steam and water, that is, if there is water in the vessel in which the steam expands, the relation between volume and pressure during expansion depends on the proportion of water in the mixture. As long as there is water present, the steam will be saturated and the pressure during isothermal expansion will be constant. This will be evident if we consider the fact that the pressure of saturated steam (steam in contact with water) depends solely on the temperature; if the temperature is constant, the pressure must also be constant, no matter what the volume may be. During isothermal expansion, the heat that is added merely changes some of the water to vapor, which fills the increased space, and there is no change in the pressure of the original steam.

If water is present during the adiabatic expansion of steam, it will give up some of its heat to assist the steam in doing

its work; in consequence of the heat derived from the water, the temperature and pressure of the steam will fall slower during expansion as the quantity of water from which heat can be derived is greater.

18. Expansion Curve of Steam.—A consideration of the above outline of the effect of water on the expansive action of steam will make it clear that an innumerable variety of curves, depending on the quantity of water present and the conditions under which expansion takes place, will correctly represent the relation between the pressures and volumes for the expansion of saturated steam. It has, however, been found that under the conditions generally existing in the cylinder of a steam engine, the curve that most nearly represents the relation between pressures and volumes is the **equilateral hyperbola,** which is the curve that shows the relation between the pressures and the volumes of a perfect gas when it expands according to Mariotte's law.

CALCULATING THE WORK DONE ON A MOVING PISTON.

19. Net or Effective Pressure.—A piston that is being pushed through a cylinder by the expansive force of a gas or vapor acting on one side must generally overcome the resisting force of a gas or vapor on its opposite side. Thus, in Fig. 3, let the space in the cylinder at the left of the piston be in communication with the steam space of a boiler in which there is an absolute pressure of 100 pounds per square inch, while the space at the right is open to the atmosphere and, in consequence, is filled with vapor at a pressure of about 14.7 pounds

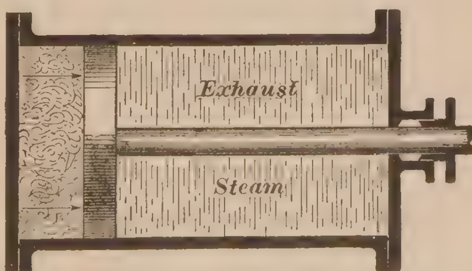


FIG. 3.

per square inch. It is evident that, neglecting the friction of the piston in the cylinder, the force that can be transmitted through the piston rod, and so made do work in overcoming some outside resistance, is the difference between the total pressure of the steam on the left and the total pressure of the air on the right of the piston. This difference is called the **net** or **effective pressure** on the piston. Since the pressure of the atmosphere is 14.7 pounds per square inch, and if the area of the piston is 100 square inches and the absolute pressure of the steam 100 pounds per square inch, the net pressure on the piston in Fig. 3 is $(100 - 14.7) \times 100 = 8{,}530$ pounds.

20. Rule for Calculating Work When Net Pressure and Piston Displacement Are Known.—The work done as the piston moves from one end of the cylinder to the other may be found as follows:

Let P = the net pressure per square foot exerted on the piston;
A = area of piston in square feet;
L = distance in feet moved over by the piston.

Then, the total net pressure on the piston is $P \times A$ pounds, and the distance through which this pressure acts is L feet. The work done is the force multiplied by the distance, or $PA \times L = PAL$ foot-pounds. But AL equals the area of the piston multiplied by the length of the stroke, which equals the volume displaced by the piston during its movement from one end of the cylinder to the other. Let V represent this volume expressed in cubic feet. Then, letting W represent the work in foot-pounds, we have $W = PAL = PV$.

It is usually more convenient to express pressures in pounds per square inch instead of pounds per square foot. Let p represent the net pressure on the piston in pounds per square inch.

Then, $$P = 144\,p,$$
and $$W = PV = 144\,p\,V.$$

Rule 1.—*To find the work done by a piston moving in a cylinder, multiply 144 by the net pressure on the piston in pounds per square inch and by the volume displaced by the piston expressed in cubic feet. The result will be the work in foot-pounds.*

The same result will be obtained by multiplying the pressure in pounds per square inch by the volume displaced by the piston in cubic inches and dividing the result by 12.

The volume displaced by a piston during a single stroke or a given period of time is often called the piston displacement for the stroke or the given period.

Example.—The piston of an engine is acted upon by a net pressure of $32\frac{1}{2}$ pounds per square inch. The volume swept through by the piston at each stroke is $5\frac{1}{2}$ cubic feet. (*a*) How much work is done at each stroke? (*b*) If the engine makes 80 strokes per minute, what horsepower does it develop?

Solution.— (*a*) According to rule **1,** the work

$$W = 144 \times 32\tfrac{1}{2} \times 5\tfrac{1}{2} = 25{,}740 \text{ ft.-lb.} \quad \text{Ans.}$$

(*b*) The number of foot-pounds per minute is $25{,}740 \times 80$, and the horsepower developed is, therefore,

$$\frac{25{,}740 \times 80}{33{,}000} = 62.4 \text{ H. P.} \quad \text{Ans.}$$

21. Work Diagrams.—The work done by a moving piston may be represented by a diagram similar to the diagrams used to represent the relation between the volumes and pressures of an expanding gas or vapor. For example, in Fig. 4, two lines OX and OY are drawn at right angles, the line OX being horizontal and the line OY vertical. Suppose that the area of the piston is 2 square feet and that the distance moved by it is 6 feet. Then, when the piston moves 1 foot, it displaces a volume of 2 cubic feet. On the line OX lay off a distance *O–1*, and let this distance represent a piston travel of 2 feet. Then, the distance *O–2*, which is twice *O–1*, represents a piston travel of 2×2 feet $= 4$ feet, and, similarly, the distance *O–3*, which is three times the distance *O–1*, represents a travel of 3×2 feet $= 6$ feet.

Since the piston area does not change, the volume swept through is proportional to the piston travel; therefore, *O–1* may be taken to represent the displacement when the

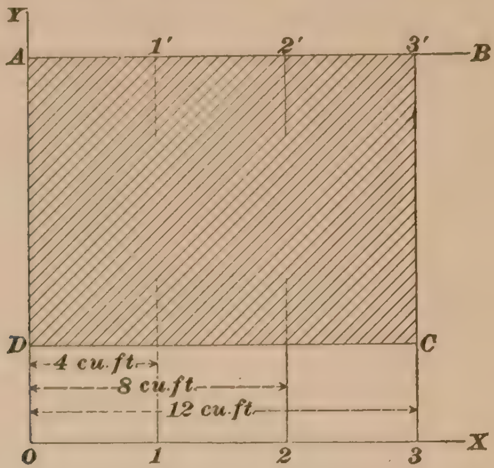


FIG. 4.

piston has traveled 2 feet. That is, *O–1* represents a volume of $2 \times 2 = 4$ cubic feet, *O–2* represents 8 cubic feet, and *O–3*, 12 cubic feet. The piston is supposed to be moving from left to right, that is, in the direction *O X*.

When the piston is at the beginning of its travel, that is, at the position represented by *O Y*, lay off on the line *O Y* a distance *O A*, which, to the scale selected, represents the pressure on the left side of the piston. Suppose the pressure is 60 pounds per square inch. Then, if *O A* is 2 inches, the scale is $\frac{60}{2} = 30$ pounds; that is, a vertical height of 1 inch represents 30 pounds per square inch pressure. Suppose the pressure to be the same throughout the stroke. Then, when the piston is at the point represented by *1*, the pressure is represented by the distance *1–1′*, which is equal to *O A*. Likewise, when the piston is in positions *2* and *3*, the distances *2–2′* and *3–3′*, respectively, represent the pressures at those points. In brief, the pressure upon the left side of the piston at any position may be found by measuring the

vertical distance between the lines $O\,X$ and $A\,B$ at that point and multiplying by the scale, 30 pounds per inch of height. In a similar manner, lay off on the line $O\,Y$ a distance $O\,D$, which, to the scale already used, represents the pressure of the atmosphere on the right of the piston, and is, therefore, equal to $\frac{14.7}{30} = .49$ inch. Since this pressure on the right of the cylinder is constant throughout the stroke, the distance from any point on the line $O\,X$ to the line $D\,C$ parallel to $O\,X$ represents the opposing pressure on the piston when it is at the corresponding point of its stroke.

22. The net pressure on the piston is represented by the distance $D\,A$ $(= O\,A - O\,D)$. We have shown that, to the scale selected, O–$3 = D\,C$ represents the piston displacement. According to rule **1**, the work done by the piston is proportional to the net pressure multiplied by the volume. Now, on the diagram of Fig. 4, $D\,A$ represents the net pressure and $D\,C$ the volume. But $D\,A \times D\,C =$ area $A\,3'\,C\,D$. Hence, the area $A\,3'\,C\,D$ must, to some scale, represent the work done by the piston.

$A\,O$ is 2 inches and $O\,D$ .49 inch; hence the distance $D\,A = O\,A - O\,D$ is $2 - .49 = 1.51$ inches; $D\,C$ equals 2 inches. Therefore, the area of the diagram is $1.51 \times 2 = 3.02$ square inches. The scale of pressure adopted was 1 inch equals 30 pounds per square inch. Hence, $p = 30 \times D\,A$. Since $D\,C$ $(= 2$ inches) represents 12 cubic feet of volume, the scale of volumes must be $\frac{12}{2} = 6$ cubic feet per inch of length. Hence, $V = 6 \times D\,C$. Then, from rule **1**, the work is

$$\begin{aligned} W &= 144\,p\,V = 144 \times (30 \times D\,A) \times (6 \times D\,C) \\ &= 144 \times 30 \times 6 \times (D\,A \times D\,C) \\ &= 144 \times 30 \times 6 \times 3.02 = 78{,}278.4 \text{ foot-pounds.} \end{aligned}$$

23. The diagram may be used in another way. The distances O–1, O–2, and O–3 may represent the distances moved through by the piston instead of the volumes displaced by it. Then, $D\,C$ represents the stroke of the piston, in this case 6 feet, and since $D\,C = 2$ inches, the horizontal scale is $\frac{6}{2} = 3$ feet of piston travel $= 1$ inch of length.

The work is

$$W = 144\, p\, A\, L.$$

As before,

$$p = 30 \times DA,$$
$$L = 3 \times DC,$$

and $$A = 2 \text{ square feet.}$$

Hence, $$W = 144 \times (30 \times DA) \times 2 \times (3 \times DC),$$
$$= 144 \times 30 \times 2 \times 3 \times (AD \times DC)$$
$$= 78{,}278.4 \text{ foot-pounds.}$$

The latter method is the one usually employed in calculating the horsepower of an engine by means of the indicator diagram.

24. Diagrams for Varying Pressures.—The diagram of Fig. 4 is very simple, because the pressure on both sides of the piston is constant throughout the stroke, thus making the diagram a rectangle. Suppose the pressure decreases uniformly throughout the stroke, as shown in Fig. 5. Here the net pressure at the beginning of the stroke is represented by the distance DA, and at the end of the stroke by the distance CB. To calculate the work, it is necessary to find the average net pressure throughout the stroke. In this case the diagram is a trapezoid; the average pressure is, therefore, represented by the line $HG = \frac{1}{2}(DA + CB)$. This distance HG is called the **mean ordinate** of the diagram $ABCD$. It has such a length that, being multiplied by the distance DC, it will give the area of a rectangle $EFCD$ that will be equal to the original area $ABCD$. The work is found by multiplying this mean ordinate by the length DC, then by the scales of pressures and volumes, and by 144.

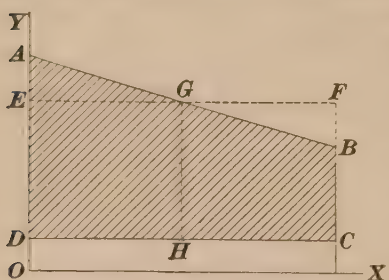


FIG. 5.

25. In Fig. 6 diagrams taken from both sides of the piston of an actual steam engine are shown on the same

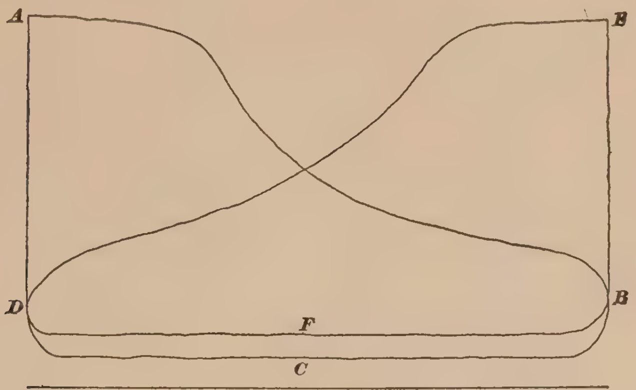


Fig. 6.

card. The line $A\ B$ represents the varying steam pressure during the forward stroke and the line $B\ C\ D$ of the crank-end diagram represents the back pressure opposing the motion of the piston during the forward stroke. Hence, the net pressure at any piston position is given by the vertical distance between the line $A\ B$ of one diagram and the line $D\ C\ B$ of the other diagram at the point representing the piston position. Likewise, the net pressure for any point of the return stroke is given by the vertical distance between the line $E\ D$ of the crank-end diagram and the back-pressure line $B\ F\ D$ of the head-end diagram. The net work done by the piston, as in the preceding cases, is given per stroke by the area $A\ B\ C\ D$ for the forward stroke and the area $E\ D\ F\ B$ for the return stroke. It will be noticed that the area $B\ C\ D\ F$ has been taken from one diagram and added to the other diagram.

Now, to find the average work per stroke of a double-acting engine, the sum of the areas representing the work done during the forward and return stroke is divided by 2. Evidently, the sum of the areas will be the same whether we add the areas $A\ B\ C\ D$ and $E\ D\ F\ B$ or add the areas of

each diagram, as *A B F D* and *E D C B*. Hence, the *average work* will be correctly given by considering the area of each diagram as representing the work done on the side of the piston the diagram was taken from and dividing the sum of the areas by 2. While the assumption that the area of each diagram represents the net work done on its side of the piston of a double-acting engine is not entirely correct, it is, nevertheless, a very convenient assumption to make, and will *not* cause any error in finding the *average pressure* per stroke when both diagrams are considered. The convenience of making the assumption just explained is best exemplified in case of diagrams taken on separate cards; in that case it would be necessary to very carefully transfer the back-pressure lines from one card to the other in order to get the correct area representing the work done on each side of the piston. This is a tedious operation calling for considerable skill in the use of drawing instruments; the necessity for this operation is obviated by making the assumption stated.

In a single-acting engine, which takes steam on one side of the piston only, the other side of the piston being open to

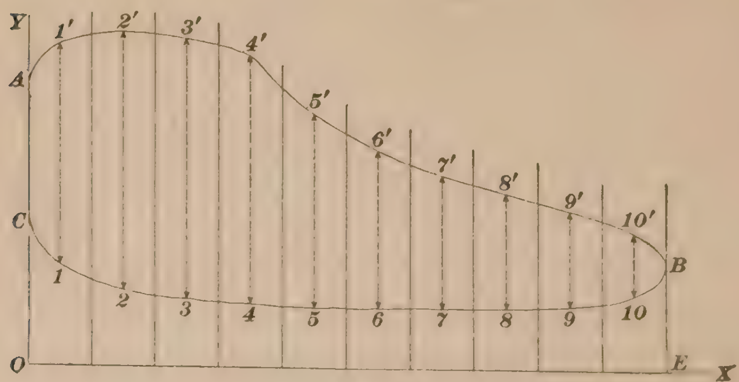


FIG. 7.

the atmosphere, the area of the diagram represents correctly the work done during the revolution. This can readily be seen by a consideration of Fig. 7, where *O E* is the

atmospheric line. The work done by the piston during the forward stroke is given by the area $O A B E$, and the work that must be done on the return stroke to make the piston return to the beginning of the working stroke is given by the area $O C B E$. Hence, the net work is equal to the difference of the areas $O A B E$ and $O C B E$, which is the area $A B C$ (the area of the diagram).

26. To find the area of an indicator diagram, we must find its mean ordinate. This may be done approximately in the following manner: Divide the length $O E$ of the diagram (see Fig. 7) into a number of equal parts (10 or 20 parts are most convenient) and through each division draw a vertical line. Half way between these vertical lines draw the lines 1–$1'$, 2–$2'$, 3–$3'$, etc., extending between the lines $A B$ and $B C$. These vertical distances between the two curves are called **ordinates.** As shown in the figure, there are ten of these ordinates equally distant from each other. If their lengths are all added together and the sum divided by the number of ordinates, the result is the average distance between the lines, or the mean ordinate.

This ordinate multiplied by the distance $O E$ gives the area of the diagram. Usually both the ordinate and $O E$ will be measured in inches; the area will then be expressed in square inches. The area being found, the work is calculated by rule **1.** That is, multiply the area by the vertical scale of pressures, by the horizontal scale of volumes, and by 144. The result is the work in foot-pounds.

Example.—The area of a diagram like that shown in Fig. 7 is found to be 7.34 square inches. The vertical scale of pressure is 36 pounds equals 1 inch, and the horizontal scale of volumes is $2\frac{1}{4}$ cubic feet equals 1 inch. What is the work per stroke of piston?

Solution.—Multiply the area by the horizontal and vertical scales, and by 144, or work $= 7.34 \times 36 \times 2\frac{1}{4} \times 144 = 95{,}126.4$ ft.-lb. Ans.

27. Work Diagram for Expanding Steam.—In connection with the diagram of Fig. 4, the piston area was taken as 2 square feet and the length of stroke as 6 feet. Fig. 8 shows the pressure diagram on the supposition that steam from the boiler is shut off when the piston has

reached one-third its stroke. Up to that point steam has entered from the boiler at a constant pressure, shown by the line AB. The volume of steam in the cylinder at this point is 4 cubic feet. As the piston moves forwards, the pressure begins to fall. When two-thirds the stroke is completed, the steam that previously occupied 4 cubic feet now occupies 8 cubic feet, that is, its volume is doubled, and assuming the expansion to follow Mariotte's law, its pressure should be one-half what it was originally, or $\frac{1}{2}\,bB$. When the piston reaches the end of the stroke, the steam occupies 12 cubic feet, or three times its original volume. Therefore,

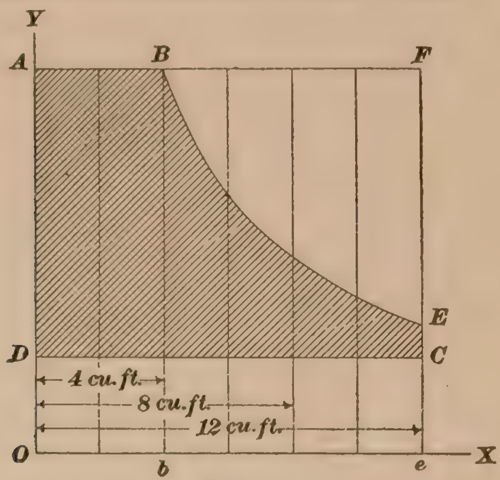


FIG. 8.

its pressure is one-third the original pressure; that is, $eE = \frac{1}{3}\,bB$. The line BE shows the fall of pressure during the last two-thirds of the stroke. When steam is shut off from the boiler in this manner and does work at the expense of its own heat, it is said to be used expansively.

28. Gain in Work by Using Steam Expansively.—It was found that the area of the diagram of Fig. 4, which represents the work done when the steam followed the piston full stroke, was 3.02 square inches, and the work done per stroke of piston was 78,278.4 foot-pounds. The area of the diagram $ABECD$, Fig. 8, which represents the work done

when the steam supply to the cylinder was cut off when one-third of the piston stroke was completed, is nearly 1.82 square inches. The work done per stroke is, therefore, $1.82 \times 30 \times 6 \times 144 = 47,174.4$ foot-pounds.

In the first case, a cylinder full of steam, 12 cubic feet, was taken from the boiler, and the work obtained from each cubic foot was, therefore, $\frac{78,278.4}{12} = 6,523.2$ foot-pounds. In the second case only 4 cubic feet of steam was taken from the boiler. Consequently, the work done by each cubic foot of steam used was $\frac{47,174.4}{4} = 11,793.6$ foot-pounds, or nearly twice as much as was done by a cubic foot when the steam followed the piston for the full stroke.

EXAMPLES FOR PRACTICE.

1. The mean ordinate of a diagram similar to that shown in Fig. 7 is 1.2 inches long. The vertical scale of pressure is 1 inch = 40 pounds per square inch, and the horizontal scale of distances is 1 inch = 10 inches. The length of the diagram is 3 inches, and 1 foot of actual length of the vessel that contains the steam represents a volume of 452 cubic inches. What is the work done in one stroke of the piston? Ans. 4,520 ft.-lb.

2. The mean ordinate of a diagram is .89 inch; the length of the diagram, 3.2 inches; the vertical scale of pressures, 1 inch = 50 pounds per square inch; the horizontal scale of volumes, 1 inch (diagram) = .56 cubic foot. Find the work done in 12 strokes.
Ans. 137,797.6 ft.-lb.

HORSEPOWER OF STEAM ENGINES.

INDICATED HORSEPOWER AND NET HORSEPOWER.

29. The relation between the pressures on the two sides of a moving piston and the work done on the piston was explained in Arts. **19** to **26**, and the student is advised to carefully review the explanation there given in conjunction with his study of this section. When the work done in a

given period of time is known, the corresponding horsepower is obtained as follows: *Having the work given in foot-pounds per minute, to find the horsepower divide by 33,000; if the work is given in foot-pounds per second, the horsepower is found by dividing by 550.* Horsepower is often abbreviated to H. P.

30. Indicated Horsepower.—The indicator furnishes the most ready method of measuring the pressures on the piston of a steam engine and, in consequence, of determining the amount of work done in the cylinder and the corresponding horsepower. The power measured by the use of the indicator is called the **indicated horsepower.** It is the total power developed by the action of the net pressures of the steam on the two sides of the moving piston. The indicated horsepower is generally represented by the initials I. H. P.

31. Friction horsepower is the part of the indicated horsepower that is absorbed in overcoming the frictional resistances of the moving parts of the engine. If the engine is running light—with no load—all the power developed in the cylinder is absorbed in keeping the engine in motion, and the friction horsepower is equal to the indicated horsepower. This principle furnishes a simple approximate method of finding the friction horsepower of a given engine; since, however, the friction between the surfaces increases with the pressure, the power absorbed in overcoming engine friction will be greater as the load on the engine is increased.

32. Net horsepower is the difference between the indicated and the friction horsepower. It is the power the engine delivers through the flywheel or shaft to the belt or the machine driven by it, and is sometimes called the **delivered horsepower.** Since the power an engine is capable of delivering when working under certain conditions is often measured by a device known as a *Prony brake*, the net horsepower is also called the **brake horsepower.**

33. The **mechanical efficiency** of an engine is the ratio of the *net horsepower* to the *indicated horsepower;* or it is the percentage of the mechanical energy developed in the cylinder that is utilized in doing useful work.

To find the efficiency of an engine, when the indicated and net horsepowers are known:

Rule 2.—*Divide the net horsepower by the indicated horsepower.*

EXAMPLE.—The indicator diagrams taken from an engine running under full load show the I. H. P. to be 238.5. The diagrams taken when the engine is running at the same speed under no load show a horsepower of 39.7. (*a*) What is the approximate net H. P. developed by the engine? (*b*) What is the efficiency of the engine?

SOLUTION.— (*a*) Approximate net H. P. = I. H. P.—friction H.P. =

$$238.5 - 39.7 = 198.8. \quad \text{Ans.}$$

(*b*) By rule **2**, the efficiency is

$$\frac{198.8}{238.5} = .834 = 83.4 \text{ per cent.} \quad \text{Ans.}$$

The mechanical efficiency of a good engine is from 75 to 90 per cent.

MEASURING THE INDICATED HORSEPOWER.

34. In accordance with the principles explained in Arts. **19** to **26**, when the net pressure on the piston and the piston displacement for a given period of time are known, the work done during the given period can be calculated. The usual period of time considered when calculating the power of an engine is 1 minute; since 33,000 foot-pounds of work per minute is equal to 1 horsepower, the horsepower is obtained by dividing the work done in one minute by 33,000.

FINDING THE M. E. P.

35. The **mean effective pressure**, or M. E. P., is defined as the average pressure urging the piston forwards during its entire stroke in one direction, less the pressure that resists its progress.

36. The mean effective pressure may be found in three ways:

1. The area of the diagram in square inches may be measured by an instrument called the *planimeter;* the M. E. P. is then found by dividing the area of the diagram in square inches by the length of the diagram in inches and multiplying by the scale of the spring.

EXAMPLE.—The area of the diagram is 4.2 square inches and the length is 3.5 inches; a 40 spring being used, find the M. E. P.

SOLUTION.— $\frac{4.2}{3.5} \times 40 = 48$ lb. per sq. in., M. E. P. Ans.

2. A special form of planimeter may be used by means of which the M. E. P. may be measured directly.

3. Where a planimeter is not available, the M. E. P. may be found with a fair degree of accuracy by multiplying the length of the mean ordinate by the scale of the spring.

37. The Planimeter.—A common form of this instrument is shown in Fig. 9. It consists of two arms hinged

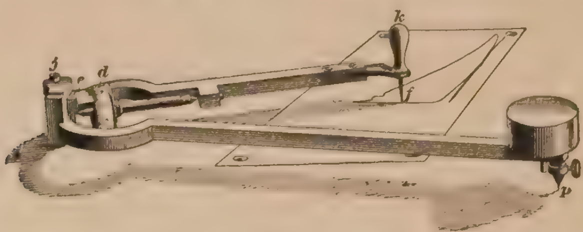


FIG. 9.

together by a pivot joint at *j*. One arm carries a recording wheel *d*, which rolls on the surface to which the card is fastened, while the outline of the diagram is being traced by the point *f*. The needle point *p* is fixed in the paper or drawing board, and remains stationary during the operation.

The indicator card should be fastened to a smooth table or drawing board that has been previously covered with a piece of heavy unglazed paper or cardboard. The point *p* should be placed far enough from the card to enable the wheel to roll on the unglazed paper without touching the card, as it will slip if rolled over a smooth surface. Set

the zero of the wheel *d* opposite the vernier *e*; then, with the tracing point *f*, follow the line of the diagram carefully, *going around the diagram in the direction of the hands of a watch*, and stop exactly at the starting point.

38. Reading the Vernier.—The area is read from the recording wheel and vernier as follows: The circumference of the wheel is divided into 10 equal spaces by long lines that are consecutively numbered from 0 to 9. Each of these spaces represents an area of 1 square inch and is subdivided into 10 equal spaces, each of which represents an area of .1 square inch. Starting with the zero line of the wheel opposite the zero line of the vernier and moving the tracing point once around the diagram, the zero of the vernier will be opposite some point on the wheel; if it happens to be directly opposite one of the division lines on the wheel, that line gives the exact area in tenths of a square inch. The zero of the vernier, however, will probably be between two of the division lines on the wheel, in which case write down the inches and tenths that are to the left of the vernier zero, and from the vernier find the nearest hundredth of a square inch as follows: Find the line of the vernier that is exactly opposite one of the lines on the wheel. The number of *spaces on the vernier* between the vernier zero and this line is the number of hundredths of a square inch to be added to the inches and tenths read from the wheel. An example is presented in Fig. 10, where the 0 of the vernier lies between the lines on the wheel representing 4.7 and 4.8 square inches, respectively, showing that the area is something more than 4.7 square inches. Looking along the vernier it is seen that there are three spaces between the vernier zero and the line of the vernier that coincides with one of the lines on the wheel; this shows that .03 square inch is to be added to the 4.7 square inches read from the wheel, making the area

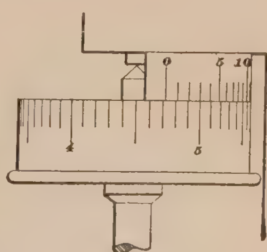


FIG. 10.

4.73 square inches, to the nearest hundredth of a square inch.

39. Modifications of Planimeter.—Planimeters are made in a number of different forms, some of which differ considerably from that shown in Fig. 9. One of the most important modifications in the general form is found in the **Lippincott** and the **Willis** planimeters. In these instruments the wheel, in addition to its rotary motion, slides in the direction of the axis of its spindle, and the area is indicated by the amount of this sliding motion as measured by a scale parallel to the axis. The **Coffin averaging instrument** is another modification, in which the end of the bar to which the wheel and vernier are attached is guided along a straight line by a slot instead of being jointed to another bar.

40. Measuring the M. E. P. Directly.—With the planimeter illustrated in Fig. 9, the M. E. P. is found by dividing the area as measured by the instrument by the length of the diagram and multiplying the quotient by the scale of the spring. Many planimeters, however, including those mentioned in the last article, can be used to measure the M. E. P. directly, no calculation being required. For this purpose, special adjustments and scales are provided by means of which the instrument can be set to correspond to the length of the diagram and the scale of the spring. The makers furnish complete instructions for the use of each of these special attachments.

41. Hints for Use of Planimeter.—It is well to so place the fixed point (*p*, Fig. 9) of the instrument that, as the tracing point moves around the diagram, the arms will swing about equally on each side of a position at right angles with each other. A slight dot is generally made with the tracing point to mark the point at which its motion around the diagram begins; when the tracing point reaches this dot in the paper, the operator knows that the motion around the diagram has been completed. The direction of motion of the tracing point must always be the same as that of the hands of a watch; motion in the opposite direction

will move the wheel in the wrong direction and give a negative reading for the area.

When measuring diagrams with loops, like Fig. 11, move the tracing point so that it will follow the outline of the loops in a direction opposite to the direction of motion of the hands of a watch, as is indicated by the arrowheads on the diagram, Fig. 11. This will cause the instrument to automatically subtract the areas of the loops from the area of the main part of the diagram.

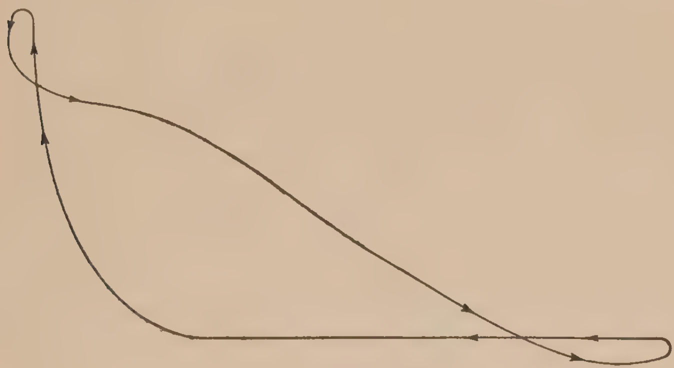


FIG. 11.

An excellent check on the work is to start with the recording wheel at zero and pass the tracing wheel around the diagram two or three times, noting the reading of the wheel each time the tracing point returns to the point of starting. Each reading of the wheel divided by the number of times the outline of the diagram has been traced should give, very nearly, the value of the first reading; if there is a considerable difference between the first reading and the value obtained by dividing the second reading by 2 or the third reading by 3, it is an indication that an error has been made, and the work should be repeated. If the difference is small, the work may be assumed to be satisfactory and the value to be used for the area or the M. E. P. may be taken as the average found by dividing the last reading by the number of times the tracing point passed around the diagram.

42. Finding the M. E. P. by Ordinates.—This operation may be performed by the aid of two triangles, a scale, and a hard lead pencil; if two triangles are not available, a single triangle and a straightedge will suffice. Lines perpendicular to the atmospheric line and tangent to the two ends of the diagram must first be drawn; the perpendicular distance between these tangents will be the length of the diagram, and this length must be divided into some number of equal parts (10 or 20 parts are the most convenient, but any other number may be used). Midway between any of the points of division draw a line parallel to the two tangents; the part of this line included between the lines of the diagram is the middle ordinate of its corresponding space. The sum of the lengths of all of these middle ordinates divided by the number of spaces is the mean ordinate and gives, approximately, the average height of the diagram. The length of the mean ordinate should agree very nearly with the value obtained by dividing the area of the diagram—as measured by a planimeter—by the length of the diagram. The M. E. P. is found by multiplying the length of the mean ordinate by the scale of the spring with which the diagram was taken.

If a scale graduated to correspond with the scale of the spring is available, the M. E. P. may be obtained by measuring the ordinates in pounds instead of in inches; the sum of the lengths of the ordinates as so measured divided by their number gives the M. E. P. of the diagram. For example, let the scale of the spring be 40, then each $\frac{1}{40}$ inch in the length of an ordinate represents a pressure of 1 pound per square inch, and by measuring the length of an ordinate with a scale graduated in fortieths of an inch, the number of pounds pressure represented by that ordinate is found.

43. A convenient method of finding the sum of the lengths of the ordinates of a diagram, and one that is especially to be recommended when a decimal scale is not available, is the following: Take a strip of paper having a

straight edge a little longer than the sum of the lengths of the ordinates. Lay this strip along the first ordinate. From the point on the strip representing one end of the first ordinate lay off the length of the next ordinate. In the same way lay off on the strip the length of each of the ordinates in succession. The length of the strip included between the extreme, or first and last, points so marked will be equal to the sum of the lengths of the ordinates, and this length divided by the number of ordinates will give the length of the *mean ordinate.*

EXAMPLE.— (*a*) The lengths between the extreme points on a strip of paper on which has been laid off successively the lengths of the 10 ordinates of an indicator diagram is $12\frac{5}{16}$ inches. What is the length of the mean ordinate to the nearest .001 inch? (*b*) The diagram was taken with a 20 spring; what is the M. E. P.?

SOLUTION.— (*a*) Reducing the fractional parts of the sum of the lengths of the ordinates to a decimal, we have $\frac{5}{16}$ inch = .3125 inch. The length of the mean ordinate is, then, $12.3125 \div 10 = 1.23125$ inches, or to the nearest .001, 1.231 inches. Ans.

(*b*) Multiplying the length of the mean ordinate by the scale of the spring, the M. E. P. is $1.231 \times 20 = 24.62$ lb. per sq. in. Ans.

44. Locating the Ordinates.—The length of the diagram will seldom be divisible into equal parts that can readily be laid off by a scale, and to divide the length into equal parts by a cut-and-try process will be found very tedious. These difficulties may, however, be overcome by an application of a simple geometrical principle, in the manner illustrated in Fig. 12. The tangent lines *a b* and *c d* are first drawn perpendicular to the atmospheric line *m n*. A scale is then selected so graduated that when the *0* mark is placed on the line *a b* and the scale lies diagonally across the diagram, the desired number of spaces will be included between the *0* mark and a mark that will fall on the line *c d*. In Fig. 12 it was desired to divide the diagram so as to get 10 ordinates. The length of the diagram is a little less than 5 inches; a scale graduated in inches can, therefore, readily be placed with the *0* mark on the line *a b* and the 5-inch mark on the line *c d*. Lines drawn parallel to *a b* and *c d*

through each of the inch and half-inch marks from *a* to *d* would evidently divide the diagram into 10 spaces of equal

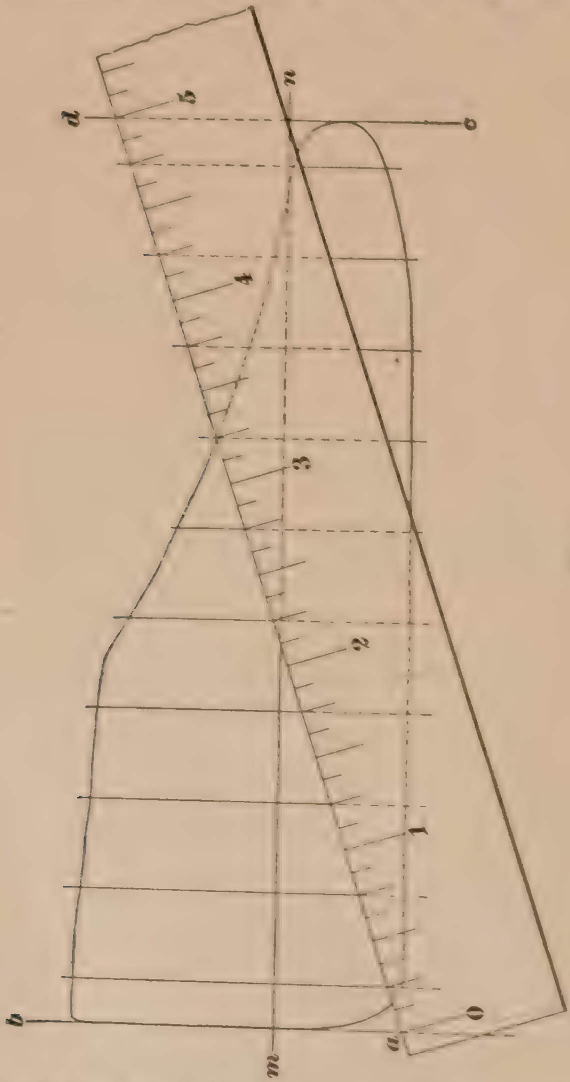


FIG. 12.

width, and since the ordinates are to be drawn through the middle of these spaces, we see that to locate the ordinates

it is only necessary to make a mark on the diagram opposite each of the quarter-inch marks on the scale, and draw parallels to *a b* and *c d* through these marks.

45. Mean Ordinate of a Diagram With Loops.—To find the mean ordinate of a diagram with loops (see Fig. 11), subtract the sum of the lengths of the ordinates of the loops from the sum of the lengths of the ordinates of the main part of the diagram and divide by the total number of ordinates. In order to get reasonably accurate results with a diagram of this kind, it will generally be necessary to use a greater number of ordinates than are required for a more simple form of diagram.

46. Approximate Determination of M. E. P.—To approximately determine the M. E. P. of an engine when the point of apparent cut-off is known, and the boiler pressure, or the pressure per square inch in the boiler, from which the supply of steam is obtained, is given, and when an indicator diagram is not obtainable, use the following rule:

Rule 3.—*Add 14.7 to the gauge pressure and multiply the number opposite the fraction indicating the point of cut-off in the table, Art.* **46,** *by the pressure. Subtract 17 from the product and multiply by .9. The result is the M. E. P. for good, simple non-condensing engines.*

Cut-off.	Constant.	Cut-off.	Constant.	Cut-off.	Constant.
$\frac{1}{6}$	.545	$\frac{3}{8}$	.773	$\frac{2}{3}$	.943
$\frac{1}{5}$	.590	.4	.794	.7	.954
$\frac{1}{4}$	.650	$\frac{1}{2}$	.864	$\frac{3}{4}$	.970
.3	.705	.6	.916	.8	.981
$\frac{1}{3}$	.737	$\frac{5}{8}$	.927	$\frac{7}{8}$	.993

If the engine is a simple condensing engine, subtract the pressure in the condenser instead of 17. The fraction

indicating the point of cut-off is obtained by dividing the distance that the piston has traveled when the steam is cut off by the whole length of the stroke; i. e., it is the apparent cut-off. For a $\frac{2}{3}$ cut-off and 92 pounds gauge pressure in the boiler, the M. E. P. is $[92 + 14.7) \times .943 - 17] \times .9 = 75.3$ pounds per square inch.

It is to be observed that this rule cannot be applied to a compound engine or any other engine in which the steam is expanded in successive stages in several cylinders.

EXAMPLE.—Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{1}{2}$ stroke and making 240 revolutions per minute. The boiler pressure is 80 pounds gauge.

SOLUTION.— $80 + 14.7 = 94.7$. Using rule **3** and table, Art. **46**, the constant for $\frac{1}{2}$ cut-off is .864, and $.864 \times$ boiler pressure $= .864 \times 94.7 = 81.82$. M. E. P. $= (81.82 - 17) \times .9 = 58.34$ lb. per sq. in. Ans.

EXAMPLES ON FINDING THE M. E. P.

EXAMPLE 1.—The projection of the head-end diagram shown in Fig. 13 upon the atmospheric line is the distance $A\ Z$, and it is divided,

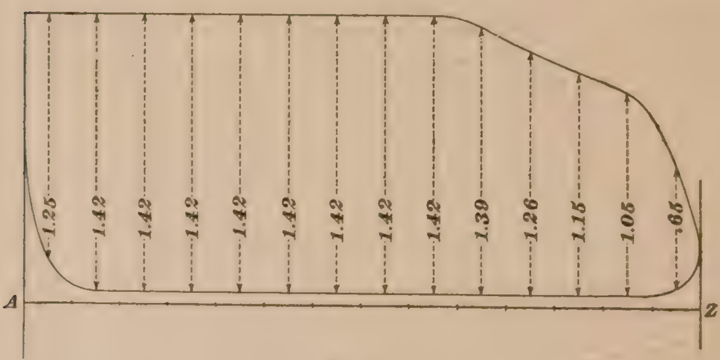


FIG. 13.

in this case into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself. The scale of the spring used in

obtaining the diagram was 40 pounds. Find (*a*) the length of the mean ordinate and (*b*) the M. E. P. of the diagram.

SOLUTION.— (*a*) The sum of the lengths of the 14 ordinates is 18.11 inches; the length of the mean ordinate is, therefore, 18.11 ÷ 14 = 1.294 in., nearly. Ans.

(*b*) Multiplying the length of the mean ordinate by the scale of the spring, we have M. E. P. = 1.294 × 40 = 51.76 lb. per sq. in. Ans.

EXAMPLE 2.—The projection of the crank-end diagram shown in Fig. 14 upon the atmospheric line is the distance *A Z*, and it is divided

FIG. 14.

in this case into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself. The scale of the spring is 40 pounds. Find (*a*) the mean ordinate and (*b*) the M. E. P. of the diagram.

SOLUTION.— (*a*) The sum of the lengths of the 14 ordinates is 17.78 inches; the length of the mean ordinate is, therefore, 17.78 ÷ 14 = 1.27 in. Ans.

(*b*) Multiplying the mean ordinate by the scale of the spring, we have M. E. P. = 1.27 × 40 = 50.8 lb. per sq. in. Ans.

EXAMPLE 3.—What was the average M. E. P. in the cylinder during the revolution represented by the two diagrams in examples 1 and 2?

SOLUTION.—Since the M. E. P. in the head end was 51.76 pounds per square inch and that in the crank end was 50.8 pounds per square inch, the average for the two strokes making up the complete revolution was $\frac{51.76 + 50.8}{2} = 51.28$ lb. per sq. in. Ans.

CALCULATING THE INDICATED HORSEPOWER.

47. General Rule for Calculating I. H. P.—Knowing the dimensions and speed of the engine and the mean effective pressure on the piston, we have all the data for finding the rate of work done in the engine cylinder expressed in horsepowers. Work is the product of force multiplied by the distance through which it acts. In the case of the engine cylinder, the total force is the M. E. P. per square inch multiplied by the area of the piston; and the distance through which the force acts in 1 minute is the distance the piston moves in 1 minute, which is equal to the number of strokes per minute multiplied by the length of the stroke.

Rule 4.—*To find the indicated horsepower developed by the engine, multiply together the M. E. P. per sq. in., the area of piston in square inches, the length of stroke in feet, and the number of strokes per minute. Divide the product by 33,000; the result will be the indicated horsepower of the engine.*

Let I. H. P. = indicated horsepower of engine;
P = M. E. P. in pounds per square inch;
A = area of piston in square inches;
L = length of stroke in feet;
N = number of working strokes per minute.

Then, the above rule may be expressed thus:

$$\text{I. H. P.} = \frac{PLAN}{33,000}.$$

In a double-acting engine the number of working strokes per minute is twice the number of revolutions per minute. For example, if a double-acting engine runs at a speed of 210 revolutions per minute there are 420 working strokes per minute. A few types of engines, however, are single-acting; that is, the steam acts on only one side of the piston. Such are the Westinghouse, the Willans, and others. In this case, only one stroke per revolution does work, and, consequently, the number of strokes per minute to be used in the above rule is the same as the number of revolutions per minute.

Unless it is specifically stated that an engine is single-acting, it is always understood, when the dimensions of an engine are given, that a double-acting engine is meant.

48. Piston Speed.—The product $L N$ of rule 4 gives the total distance traveled by the piston in 1 minute. This is called the **piston speed.** It is usual to take the stroke in inches. Then, to find the piston speed, multiply the stroke in inches by the number of strokes and divide by 12, or, letting S represent the piston speed, $S = \frac{l N}{12}$, where l is the stroke in inches. But $N = 2 R$, where R represents the number of revolutions per minute. Hence,

$$S = \frac{l N}{12} = \frac{l \times 2 R}{12} = \frac{l R}{6}.$$

Rule 5.—*To find the piston speed of an engine, multiply the stroke in inches by the number of revolutions per minute and divide the product by 6.*

Example.—An engine with a 52-inch stroke runs at a speed of 66 revolutions per minute. What is the piston speed?

Solution.—By rule 5, $S = \frac{52 \times 66}{6} = 572$ ft. per min. Ans.

The piston speeds used in modern practice are about as follows:

	Ft. per min.
Small stationary engines........	300 to 600.
Large stationary engines........	600 to 1,000.
Corliss engines.................	400 to 750.
Marine engines..................	200 to 1,200.

49. Allowance for Area of Piston Rod.—It is generally considered sufficiently accurate to take the total area of one side of the piston as the area to be used in calculating the horsepower of an engine. The effective area of one side of the piston is, however, reduced by the sectional area of the piston rod, and if it is important that the power be calculated to the greatest practicable degree of accuracy, an allowance for the area of the piston rod must be made. This is done by

taking as the piston area one-half the sum of the areas exposed to steam pressure on the two sides of the piston. Thus, if we have a piston 30 inches diameter with a 6-inch piston rod, the average area is $\frac{30^2 \times .7854 + (30^2 \times .7854 - 6^2 \times .7854)}{2}$ $= 692.72$ square inches. If the piston rod is continued past the piston so as to pass through the head-end cylinder head, i. e., if the piston has a **tail rod,** allowance must be made for the tail rod. Thus, with a piston 30 inches diameter, a piston rod 6 inches diameter, and a tail rod 5 inches diameter, the average area is $\frac{(30^2 \times .7854 - 5^2 \times .7854) + (30^2 \times .7854 - 6^2 \times .7854)}{2}$ $= 682.9$ square inches.

EXAMPLE 1.—The diameter of the piston of an engine is 10 inches and the length of stroke 15 inches. It makes 250 revolutions per minute with an M. E. P. of 40 pounds per square inch. What is the horsepower?

SOLUTION.—The number of working strokes is $250 \times 2 = 500$ per minute. Applying rule **4**, we get

$$\text{I. H. P.} = \frac{40 \times \frac{15}{12} \times (10^2 \times .7854) \times 500}{33{,}000} = 59.5 \text{ H. P.} \quad \text{Ans.}$$

EXAMPLE 2.—In Fig. 15 are shown two indicator diagrams taken from an 18" × 20" engine, making 200 revolutions per minute. The

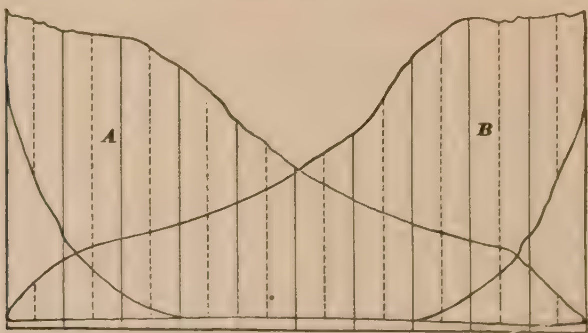


FIG. 15.

scale of the spring is 60. Compute (*a*) the mean effective pressure and (*b*) the horsepower.

SOLUTION.—(*a*) Divide the diagrams into 10 equal parts, as shown by the full lines. Then, as previously directed, draw lines or ordinates (see dotted lines in cut) perpendicular to the atmospheric line through the middle points of each of the 10 equal divisions. Measuring the lengths of all the dotted lines and adding them together, we find the sum of the lengths for diagram *A* is 7.8 inches and for diagram *B* 7.84 inches. Dividing each of these results by 10 and multiplying by the scale of the spring, we have $\frac{7.8}{10} \times 60 = 46.8$ pounds per square inch = M. E. P. for diagram *A*, and $\frac{7.84}{10} \times 60 = 47.04$ pounds per square inch = M. E. P. for diagram *B*. The average M. E. P. for both cards is $\frac{46.8 + 47.04}{2} = 46.92$ lb. per sq. in. Ans.

(*b*) To find the horsepower, the value for the M. E. P. must be substituted for *P* in the formula corresponding to rule **4**, $\frac{P\,L\,A\,N}{33{,}000} = \text{H. P.}$ Reducing the stroke to feet, and substituting the values of *P*, *L*, *A*, and *N*, we have

$$\frac{46.92 \times \frac{20}{12} \times (18^2 \times .7854) \times (200 \times 2)}{33{,}000} = 241.21 \text{ H. P.} \quad \text{Ans.}$$

ENGINE CONSTANTS.

50. An **engine constant** for a given engine is a number obtained by combining into a single factor all the factors of the horsepower rule that are constant for that engine. This factor may then be substituted for the factors that were combined to produce it, and a new rule obtained for that engine, in which the number of unknown quantities is less than in the original rule. The labor involved in calculating the I. H. P. for the engine is thus considerably reduced.

51. Constant for a Uniform Speed of Rotation.—When the speed of rotation of a given engine is uniform, all the factors except the mean effective pressure are constant; the engine constant for this case can, therefore, be found by the following rule:

Rule 6.—*Multiply together the length of the stroke in feet, the area of the piston in square inches, and the number of*

working strokes per minute, and divide the product by 33,000; the quotient will be the engine constant.

This rule may be expressed by the formula

$$C_u = \frac{L\,A\,N}{33,000};$$

in which C_u is the engine constant for the uniform speed of rotation, and L, A, and N have the same meaning as in the formula corresponding to rule **4**. The constant C_u is the horsepower of the engine for a mean effective pressure of 1 pound per square inch.

To find the I. H. P. when the engine constant for a uniform speed of rotation is known, multiply the engine constant by the M. E. P.

EXAMPLE 1.—What is the engine constant for a 16″ × 20″ engine running at a uniform speed of 200 R. P. M.?

SOLUTION.—The length L of the stroke is $\frac{20}{12}$ feet, the area A of the piston is $16^2 \times .7854 = 201$ square inches, and the number of strokes N is $2 \times 200 = 400$. Substituting these values in the formula corresponding to rule **6**, we have

$$C_u = \frac{\frac{20}{12} \times 201 \times 400}{33,000} = 4.06. \quad \text{Ans.}$$

EXAMPLE 2.—What is the I. H. P. of the engine of example 1 when the average M. E. P. for a pair of indicator diagrams is 43.2 pounds per square inch?

SOLUTION.—Multiplying the engine constant by the M. E. P., we have I. H. P. $= 4.06 \times 43.2 = 175.39$. Ans.

52. Constant for a Varying Speed of Rotation.—When the speed of rotation is variable, the engine constant is given by the following rule:

Rule 7.—*Multiply together twice the length of stroke in feet and the area of the piston in square inches; divide the product by 33,000 for a double-acting engine. For a single-acting engine, multiply the length of stroke in feet by the area of the piston in square inches and divide the product by 33,000.*

Or, $C_v = \dfrac{2\,L\,A}{33,000}$ for double-acting engines,

and $$C_v = \frac{L A}{33{,}000} \text{ for single-acting engines,}$$

where $$C_v = \text{engine constant.}$$

The value of C_v derived from these formulas is the I. H. P. of the engine for a speed of 1 revolution per minute and a mean effective pressure of 1 pound per square inch. To find the I. H. P., multiply this constant by the number of revolutions per minute and the product so obtained by the M. E. P.

53. Formulas for M. E. P. and I. H. P. in Terms of Area of Diagram.—The fact that the M. E. P. of a diagram is equal to its area in square inches divided by its length in inches and this quotient multiplied by the scale of the spring enables us to develop a formula by means of which the horsepower can be calculated from the area and length of the diagram and a constant that is obtained by multiplying the engine constant by the scale of the spring. Such a formula will be found convenient when the area of the diagram is measured by a planimeter that cannot be set to measure the M. E. P. of the diagram directly.

Let a = area of diagram in square inches;
l = length of diagram in inches;
s = scale of spring.

Then $$\text{M. E. P.} = \frac{a s}{l}.$$

This value of M. E. P. can be substituted for P in the formula corresponding to rule **4**, giving us the formula

$$\text{I. H. P.} = \frac{\frac{a s}{l} L A N}{33{,}000}.$$

For a given engine from which a number of diagrams are to be taken, the factors s, L, A, and N will generally be constant; these factors may, therefore, be combined with the factor 33,000 in the same manner as was done in Art. **51**; a constant which we will call C_a may thus be obtained which will be given by

Rule 8.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide the product by 33,000.*

Or,
$$C_a = \frac{s\,L\,A\,N}{33{,}000}.$$

The indicated horsepower will then be given by multiplying this constant by the area of the diagram if the engine is single-acting, or the average area of the two diagrams if the engine is double-acting, and dividing the product by the length of the diagram.

54. If the indicator reducing motion is so constructed that the length l of the diagrams is constant, the constant may be made to include this factor. Calling such a constant C_l, we have

Rule 9.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide this product by the product of 33,000 and the length of the diagram.*

Or,
$$C_l = \frac{s\,L\,A\,N}{33{,}000\,l}.$$

With this constant, the indicated horsepower can be found by multiplying it by the area of the diagram if the engine is single-acting, or the average area of the two diagrams when the engine is double-acting.

Example.—Calculate the value of the constant by which to multiply the area of the diagrams to find the I. H. P. for a 12″ × 16″ engine running at 250 R. P. M. when the scale of the spring is 50 and the length of the diagrams is 3½ inches.

Solution.—The length L of the stroke is $\frac{16}{12}$ feet, the area A of the piston is $12^2 \times .7854 = 113.1$ square inches, and the number N of working strokes is $2 \times 250 = 500$ per minute. Substituting these values and the given values for the scale of the spring and the length of the diagram in rule **9**, we have

$$C_l = \frac{50 \times \frac{16}{12} \times 113.1 \times 500}{33{,}000 \times 3\tfrac{1}{2}} = 32.64. \quad \text{Ans.}$$

55. Formula for I. H. P. in Terms of Total Length of Ordinates.—

Let n = number of ordinates drawn on diagram;
o = sum of the lengths of ordinates in inches;
h = length of mean ordinate in inches;
C_o = constant for calculating the I. H. P. from the ordinates;
s = scale of indicator spring.

In accordance with Art. **43**, the length of the mean ordinate is equal to the sum of the lengths of the ordinates divided by their number; that is,

$$h = \frac{o}{n};$$

and in accordance with Art. **42**, the mean effective pressure is equal to the length of the mean ordinate multiplied by the scale of the spring, or

$$\text{M. E. P.} = s\,h = s\frac{o}{n}.$$

Substituting this value of the M. E. P. for P in the formula corresponding to rule **4**, we have

$$\text{I. H. P.} = \frac{s\frac{o}{n}L\,A\,N}{33{,}000}.$$

56. For the diagrams taken from a given engine running at a uniform rate of speed, the factors s, n, L, A, and N are generally constant. They may, therefore, be combined with the constant factor 33,000 to form a new constant whose value is given by the following rule:

Rule 10.—*Multiply together the scale of the indicator spring, the length of stroke in feet, the area of the piston in square inches, and the number of working strokes per minute. Divide this product by the product of 33,000 and the number of ordinates.*

Or, $$C_o = \frac{s\,L\,A\,N}{33{,}000\,n}.$$

This constant multiplied by the sum o of the lengths of the ordinates in inches for a diagram gives the indicated horsepower for a single-acting engine. For a double-acting engine one-half the sum of the lengths of the ordinates of the two diagrams is to be taken. It is to be observed that the number of ordinates must be the same for each diagram, and that in case of a double-acting engine the *sum* of the number of ordinates of the two diagrams *must not be used.*

EXAMPLE 1.—Calculate the value of the constant C_o for diagrams taken with a 40 spring from a 28″ × 42″ engine running at 90 R. P. M. when the number of ordinates is 20.

SOLUTION.—The area A of the piston is $28^2 \times .7854 = 615.75$ square inches; the length L of the stroke is $\frac{42}{12} = 3\frac{1}{2}$ feet, and the number N of working strokes is $2 \times 90 = 180$ per minute. Substituting these and the values given for the scale s of the spring and the number n of ordinates in rule **10**, we have

$$C_o = \frac{40 \times 3\frac{1}{2} \times 615.75 \times 180}{33{,}000 \times 20} = 23.51. \quad \text{Ans.}$$

EXAMPLE 2.—What is the I. H. P. of the engine of example 1, when one-half the sum of the lengths of the 20 ordinates of the two diagrams is $19\frac{3}{16}$ inches?

SOLUTION.—

$$\text{I. H. P.} = 23.51 \times 19\tfrac{3}{16} = 451.1, \text{ nearly.} \quad \text{Ans.}$$

BRAKE HORSEPOWER.

DYNAMOMETERS.

57. Dynamometers are instruments for measuring power. They are divided into two main classes: *absorption dynamometers* and *transmission dynamometers.*

58. The most common form of **absorption dynamometer** is the Prony brake, which consists simply of a friction brake designed to absorb in friction and measure the work done by a motor, or the power given out by a shaft.

59. A **transmission dynamometer** is used to measure the power required to drive a machine or do other work; then, to determine the power required to run the shafting in a mill, a transmission dynamometer would be interposed between the shafting and the source of power, and by suitable belt connections the shafting would be driven through the dynamomoter, from which the power could be determined. Since transmission dynamometers do not enter into the work of the steam engineer, they will not be treated of here.

60. **Brake horsepower** is a term often applied to the power measured by a Prony brake or other type of absorption dynamometer. The brake horsepower of an engine or other motor working under given conditions is the same as the net horsepower. Since the power measured by an absorption dynamometer is the power a motor delivers at the shaft or flywheel, it is sometimes called the **delivered horsepower.**

61. **The Prony Brake.**—Fig. 16 represents a simple and common form of Prony brake. It consists of two wooden

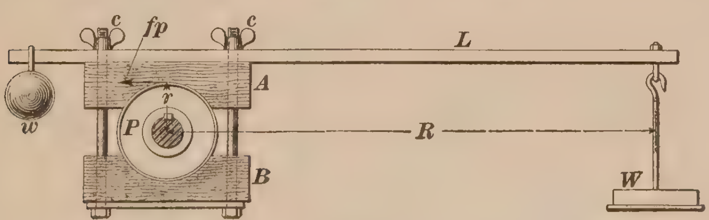


FIG. 16.

blocks A and B that are clamped together upon a pulley P by the bolts and thumbnuts c, c. The same bolts clamp an arm L to the upper block, from which a scale pan bearing a known weight W is suspended. The distance R from the center of the pulley to the perpendicular through the point from which the scale pan is suspended is also known. The counterweight w should be so adjusted as to just balance the extra length of L on the right and the weight of the scale pan.

Suppose the pulley to revolve left-handed and the bolts c, c tightened until, with a weight W in the scale pan, the lever L will remain stationary in a horizontal position. Then the foot-pounds of work absorbed by the brake can be found by multiplying the weight W by the circumference of a circle whose radius is R (in feet) and by the number of revolutions of the pulley.

Rule 11.—*To find the horsepower, multiply the weight in the scale pan by the length in feet of the lever arm about the center of the shaft, by the number of revolutions of the pulley per minute, and by 6.2832. Divide the product by 33,000.*

Or, $$\text{H. P.} = \frac{W R N \times 6.2832}{33{,}000},$$

where H. P. = number of horsepower absorbed;
R = length in feet of lever arm about center of shaft;
W = weight in scale pan;
N = number of revolutions per minute.

EXAMPLE.—A brake with an arm R 6 feet long was placed on the flywheel of an engine. If the engine ran at 200 revolutions per minute, what power did it develop when the brake balanced with 14 pounds in the scale pan?

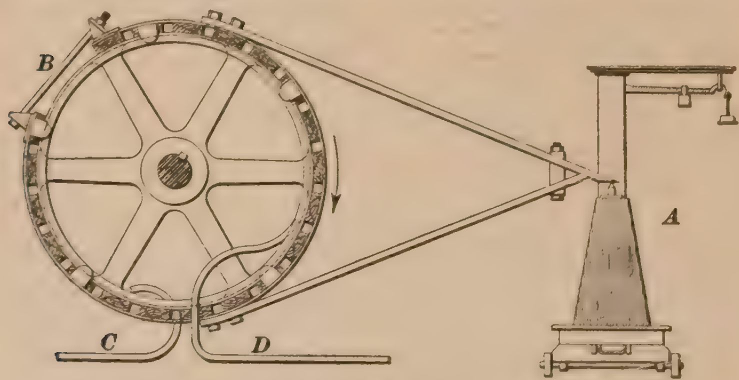


FIG. 17.

SOLUTION.—Applying the rule just given, we get

$$\text{H. P.} = \frac{14 \times 6 \times 200 \times 6.2832}{33{,}000} = 3.198. \quad \text{Ans.}$$

62. Brakes are often constructed of a metal band that extends entirely around the pulley, the rubbing surface being formed of blocks of wood fitted to the inside of the band. A weight arm is attached to one side of the pulley, and the friction is varied by means of a bolt and nut used to connect the two ends of the band.

Instead of hanging weights in the scale pan, the friction may be weighed on a platform scale, as shown in Fig. 17. In this case, the direction of rotation of both pulley and arm is the same. Rule **11** may be used for calculating the brake horsepower, substituting the weight indicated by the scale

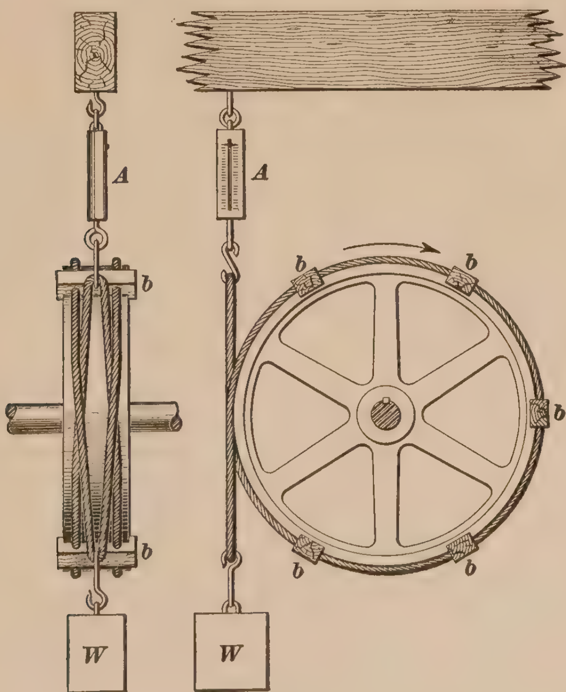


FIG. 18.

for the weight in the scale pan, and taking the length of the lever as the distance between the center of the shaft and the point where the lever presses on the platform. In

reading the weight off the scale beam, it must be remembered that the weight to be used in the calculation is the *difference* between the weight at which the scale balances when the brake is not applied and when applied.

63. It is essential that Prony brakes should be well lubricated, and for all except small powers, means must be provided for conducting away the heat generated by friction. If there are internal flanges on the brake wheel, water can be run on the inside of the rim, the flanges serving to retain the water at the sides and centrifugal force to keep it in contact with the rim. A funnel-shaped scoop can be used to remove the water. It should be attached to a pipe and placed so as to scoop out the water, which should flow continuously. This arrangement is shown in Fig. 17.

64. **A rope brake,** like that in Fig. 18, will give good results. The figure shows the construction so clearly that no description is necessary. To obtain the brake load, subtract the brake pull as registered by the spring balance from the weight. In this case, the lever arm is equal to the radius of the pulley plus one-half the diameter of the rope, expressed in feet.

STEAM CONSUMPTION OF SIMPLE ENGINES.

CALCULATIONS RELATING TO STEAM CONSUMPTION.

65. The indicator diagram also enables us to find approximately the amount of steam consumed by the engine. In referring to the steam consumption, it is customary to take as a unit the *steam consumed per horsepower per hour*. It is to be observed that the expressions "steam consumption" and "water consumption" when applied to a steam engine are synonymous.

Take a point *a* on the expansion line before the release (see Fig. 19); measure the pressure from the vacuum line, and from column 6 of the Steam Table find the weight of a

cubic foot at that pressure. The cubic contents of the cylinder (including the clearance) up to the point *a* multiplied by the weight per cubic foot, must give the weight of steam in the cylinder at this instant. This weight would be the steam consumed per stroke were it not for two circumstances. (1) When the fresh steam from the boiler enters the cylinder, it comes in contact with the cylinder walls, which have been cooled down by the exhaust steam. A

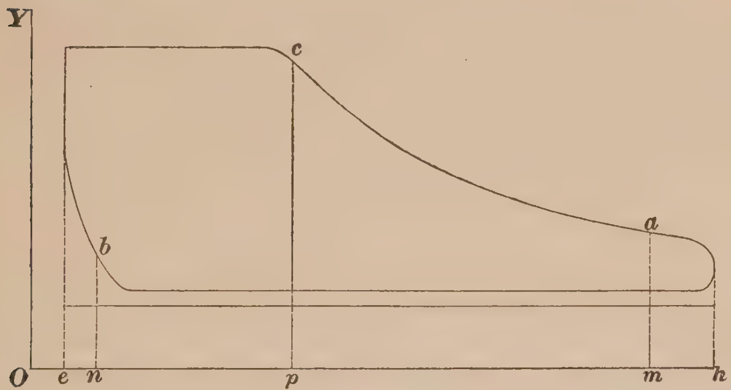


FIG. 19.

glance at the Steam Table shows that the exhaust steam is at a great deal lower temperature than the fresh steam. Consequently, part of the incoming steam condenses, and, of course, the indicator diagram takes no account of this condensed steam. Hence, the steam actually in the cylinder is less than originally entered from the boiler, because part of the original steam has changed to water. (2) On account of the closure of the exhaust port, some steam is compressed and saved.

66. To find the weight of the steam saved by compression, take a point *b* on the compression curve, measure its pressure from vacuum as before, and compute the weight of the steam in the cylinder up to *b*. Subtract this from the weight first obtained, and the difference will be the weight

of steam per stroke accounted for by the indicator. Multiply this weight per stroke by the number of strokes per hour and divide by the I. H. P. of the engine. The result will be the steam used per I. H. P. per hour.

EXAMPLE.—Fig. 19 represents an indicator diagram taken with a 45 spring from an engine having an 18″ × 24″ cylinder, running at 120 revolutions and developing 130 horsepower. The clearance is 5 per cent. Find the steam consumption per I. H. P. per hour.

SOLUTION.—Project the two ends of the diagram perpendicularly upon the vacuum line $O\,h$, as at e and h; $e\,h$ is then the length of the diagram. Lay off $e\,O$ equal to the clearance; that is, equal to 5 per cent. of $e\,h$. Draw $O\,Y$ perpendicular to $O\,h$. Take the point a near the point of release and measure the distances $a\,m$ and $O\,m$. Take the point b somewhere on the compression line and measure the distances $b\,n$ and $O\,n$. The measurements are found to be:

$$a\,m = .71 \text{ inch;}$$
$$O\,m = 3.17 \text{ inches;}$$
$$b\,n = .6 \text{ inch;}$$
$$O\,n = .333 \text{ inch.}$$

The length of the diagram $= e\,h = 3\frac{1}{3}$ inches; the length of the stroke is 2 feet. Hence, each inch of the length of the diagram equals $\frac{2}{3\frac{1}{3}} = .6$ foot of stroke. Since the scale of the indicator spring is 45, the above measurements reduced to pressures in pounds per square inch and feet of stroke become:

$$a\,m = .71 \times 45 = 31.95 \text{ pounds;}$$
$$b\,n = .6 \times 45 = 27 \text{ pounds;}$$
$$O\,m = 3.17 \times .6 = 1.9 \text{ feet;}$$
$$O\,n = .333 \times .6 = .2 \text{ foot.}$$

The area of the piston is $18^2 \times .7854 = 254.47$ square inches $= \frac{254.47}{144} = 1.767$ square feet. Consequently, the volume of steam in the cylinder when the piston is at the point represented by a is $1.9 \times 1.767 = 3.3573$ cubic feet. The volume when the piston is at b is $.2 \times 1.767 = .3534$ cubic foot. The weight of a cubic foot of steam at an absolute pressure of 31.95 pounds per square inch is found from the Steam Table to be .078723 pound; and at a pressure of 27 pounds, the weight is .067207 pound. Hence, the weight of the steam in the cylinder is $.078723 \times 3.3573 = .264297$ pound; while the weight of steam saved by compression is $.067207 \times .3534 = .023751$ pound. The steam used per stroke is, therefore, $.264297 - .023751 = .240546$ pound. To find the amount used per I. H. P. per hour, multiply the weight used per stroke

by the number of strokes per hour and divide by the I. H. P. Therefore, the required weight is

$$\frac{.240546 \times 120 \times 2 \times 60}{130} = 26.645 \text{ lb. Ans.}$$

67. Suppose the weight of the steam in the cylinder to be calculated by taking the point c near the point of cut-off. $cp = 1.59$ inches $= 1.59 \times .45 = 71.55$ pounds; $Op = 1\frac{1}{3}$ inches $= \frac{4}{3} \times .6 = .8$ foot of stroke. The volume of steam in the cylinder when the piston is at c is, therefore, $.8 \times 1.767 = 1.4136$ cubic feet. One cubic foot of steam at the pressure of 71.55 pounds, absolute, weighs .168009 pound. The weight of the steam in the cylinder is, therefore, $.168009 \times 1.4136 = .237498$ pound. Subtracting the steam saved by compression, the steam used per stroke is $.237498 - .023751 = .213747$ pound, and the steam per I. H. P. per hour is

$$\frac{.213747 \times 120 \times 2 \times 60}{130} = 23.677 \text{ pounds.}$$

Now, unless the valve leaks, the weight of the steam when the piston is at a can be no greater than when it is at c, since no fresh steam has been allowed to enter; but the calculation shows that there is .264297 pound in the cylinder when the piston is at a, and only .237498 pound when the piston is at c. This shows that $.264297 - .237498 = .026799$ pound has been condensed to water by the time the piston has arrived at c, but has been re-evaporated before the piston arrives at a. Hence, by calculating the water consumption at cut-off and then at release, a good idea of the amount of cylinder condensation may be obtained. If the steam used by the engine be actually caught and weighed and then compared with the weight as calculated from release, an idea may be obtained of the amount of condensation at release. The computed consumption is always less than the actual consumption.

68. Where there is a sufficient amount of compression, the work may be simplified by taking the two points a and b at the same height above the vacuum line, as shown in

Fig. 20. Since the absolute pressure at a and b is the same, the clearance may be left entirely out of account, and the

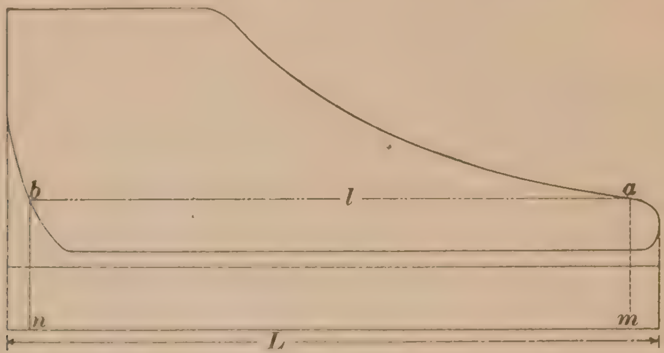


FIG. 20.

volume to be used in the computation will be $\frac{l}{L}$ times the volume of the cylinder, or, in other words, $\frac{l}{L} \times$ length of stroke × area of piston. When this method is used, the steam consumption may be found directly from the formula

$$Q = \frac{13{,}750\, l\, W}{P L},$$

in which Q is the number of pounds of steam consumed per horsepower per hour; W, the weight of a cubic foot of steam at the absolute pressure a, and P, the M. E. P.

Expressing the formula in words, we have the following rule:

Rule 12.—*Take two points, one on the expansion line and one on the compression line, both equally distant from the vacuum line. Find the pressure of the steam at these points, and from the Steam Table find the weight of a cubic foot of steam at that pressure. Multiply this weight by the distance between the two points and by 13,750. Divide the product by the M. E. P. and by the length of the diagram. The result will be the pounds of steam consumed per I. H. P. per hour, as shown by the diagram.*

EXAMPLE.—From a diagram taken from an $18\frac{1}{2}'' \times 30''$ engine, the following measurements were obtained (see Fig. 20): $a\,m = .667$ inch; $l = 3.08$ inches; $L = 3.5$ inches; M. E. P. = 35 pounds; spring, 45. What is the steam consumption per I. H. P. per hour?

SOLUTION.—The indicator diagram being taken with a 45 spring, the pressure at a is $45 \times .667 = 30$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .0742 pound. Using rule **12**,

$$Q = \frac{13,750 \times 3.08 \times .0742}{35 \times 3.5} = 25.65 \text{ lb. Ans.}$$

EXAMPLES FOR PRACTICE.

1. Size of engine, $12'' \times 20''$; length of diagram L, 3.4 inches; length l, $2\frac{1}{4}$ inches; height $a\,m$, $\frac{3}{8}$ inch; R. P. M., 230; spring, 30; M. E. P., 18 pounds per square inch. What is the steam consumption per I. H. P. per hour? Ans. 25.63 lb. per I. H. P. per hr.

2. Size of engine, $12'' \times 12''$; M. E. P., 51.1; length of diagram L, 2.6 inches; length l, 1.8 inches; height $a\,m$, .7 inch; R. P. M., 350; spring, 70. What is the steam consumption per I. H. P. per hour? Ans. 21.92 lb. per I. H. P. per hr.

3. If, in the above engine, example 2, the pressure at cut-off is 110 pounds, absolute; the clearance is 8 per cent.; the length of the diagram to the point of cut-off is .7 inch; the pressure at a point on the compression curve is 49 pounds, absolute, and the distance of this point from the end of the diagram is .14 inch, what is the steam consumption per I. H. P. per hour at cut-off? Ans. 19.44 lb. per I. H. P. per hr.

SIZE OF STEAM ENGINES.

69. The problem of selecting a size of simple engine that will develop a given indicated horsepower is capable of an infinite number of correct solutions, depending on the conditions present. The factors that determine the indicated horsepower are the mean effective pressure, the length of stroke, the diameter of the piston, and the number of revolutions per minute. Before the diameter of the piston and the length of stroke, which data constitute the size of the engine, can be determined, the boiler pressure, point of cut-off, and piston speed must be chosen. From the boiler

pressure and the point of cut-off the mean effective pressure is then estimated in the manner explained in Art. **46.** The area of the piston is then given by the following rule:

Rule 13.—*To find the piston area in square inches, multiply the indicated horsepower by 33,000 and divide by the product of the mean effective pressure and the piston speed in feet per minute.*

Or, $$A = \frac{33{,}000\,H}{PS},$$

where
A = area of piston;
H = indicated horsepower;
P = mean effective pressure;
S = piston speed.

To find the diameter, divide the result of rule **13** by .7854 and extract the square root of the quotient.

Example.—Find the piston diameter for a 25-horsepower engine using steam at 70 pounds gauge pressure, cutting off at $\frac{2}{3}$ stroke, and to have a piston speed of 300 feet per minute. Engine is non-condensing.

Solution.—By rule **3**, Art. **46**, the mean effective pressure is $[(70 + 14.7) \times .937 - 17] \times .9 = 56$ pounds per square inch. Applying rule **13**, we get

$$A = \frac{33{,}000 \times 25}{56 \times 300} = 49.1 \text{ square inches.}$$

The corresponding diameter is $\sqrt{\frac{49.1}{.7854}} = 8$ in., about. Ans.

70. Since the piston speed is the product of the number of strokes and the length of stroke, to find the latter the number of strokes must be assumed. Then, to find the length of stroke in feet, divide the piston speed by the number of strokes.

Example.—If the engine in the example given in Art. **69** is to make 120 revolutions per minute, what should be the stroke in inches?

Solution.—Stroke in feet $= \frac{300}{120 \times 2} = 1.25$ feet, or $1.25 \times 12 = 15$ in. Ans.

GOVERNORS

FORMS OF GOVERNORS

INTRODUCTION

PURPOSE OF GOVERNORS

1. The device applied to a reciprocating or rotary steam engine or to a steam turbine to maintain automatically, as closely as possible, a uniform speed of rotation of the crank-shaft or turbine shaft, is called a **speed governor,** or *governor,* for short. A device applied to a steam engine or steam turbine to automatically shut off the steam from the machine in case of accidental overspeed, or a failure of the governor that may develop a dangerous overspeed, and thereby stop the machine before harm is done, is called an *emergency governor* or a *safety governor.* A device applied to a steam pump, an air compressor, etc. to maintain a substantially uniform pressure in a vessel into which the machine discharges, is called a *pressure governor;* when applied for the purpose of maintaining a uniform level of a liquid, the device becomes a *water-level governor.*

2. There are several reasons why a steam engine or steam turbine will not run at a uniform speed unless it is governed automatically. In the first place, the load on the machine, that is, the resistance to be overcome, varies in practice, changing under some conditions almost instantaneously from no load to full load, or from full load to no load, or at least between wide limits. In the second place, in actual practice

the steam pressure, no matter what care is exercised in boiler management, cannot be maintained absolutely uniform; neither is the quality of the steam nor the degree of superheat, if superheated steam is used, the same at all times. In consequence of the lack of uniformity in the factors just mentioned, a steam engine or steam turbine, even when running under a constant load, cannot run at a uniform speed without being governed by some means. The reason for this is that the force impelling the piston of the engine against the resistance of the load, or the force exerted against the rotor of the steam turbine, varies with changes of steam pressure, steam quality, superheat, and back pressure against which the steam exhausts.

PURPOSE OF FLYWHEEL

3. In a steam turbine the force acting on the rotary element is substantially uniform during its whole revolution; in the reciprocating steam engine, however, during each revolution of the crank-shaft there are variations in the force acting on the piston that cannot be directly controlled by the governor. At the beginning of the stroke the force is much greater than it is near the end, after the pressure has been reduced by the expansion of the steam. Furthermore, the turning effect exerted on the crank by the force acting on the piston also varies. The resistance to be overcome at the circumference of the belt pulley or the resistance to be overcome by the engine shaft is generally uniform for any single revolution of the shaft. Therefore, in order to keep the speed substantially uniform during a revolution, part of the energy when the turning effect is greatest must be stored so as to be given up again when the crank is near the dead centers and the pressure on the piston has fallen below the average pressure required to do the work. This duty is performed by the flywheel. When the turning effect on the crank is greater than the average resistance, a part of the energy is absorbed in increasing the speed of the flywheel rim; when the turning effect is less than the resistance, the flywheel rim gives up a part of the energy represented by its velocity, and the speed falls. By making the diameter of

the flywheel large and the rim heavy, the variation in speed required for the energy stored and given out during a single revolution is made so small as to be unobjectionable.

The flywheel also serves as a reservoir for storing the energy represented by the work done in the cylinder when there is a sudden drop in the load on the engine that cannot be immediately taken care of by the governor, or for furnishing a supply of energy to do the work when there is a sudden increase in the load. It thus aids in preventing too great a change in speed when there is a sudden change in the load.

METHODS OF GOVERNING SPEED

4. In the reciprocating steam engine the work done in the cylinder can be regulated only by varying the average net pressure on the piston. This average pressure can be varied by three different methods: First, by varying the pressure of the steam admitted to the cylinder; second, by changing the point of cut-off so as to vary the ratio of expansion; third, by varying the back pressure or the pressure opposing the piston motion. Of these three methods, the first and second are the ones most generally used, although the third is often used in conjunction with the second in reciprocating steam engines.

In the first method, the flow of steam to the steam chest is controlled by a valve that regulates the pressure in the steam chest without affecting the expansion of the steam in the cylinder. The valve *throttles* the steam, that is, it obstructs its flow to a greater or less extent; this method of governing is, therefore, called *throttling*, and the device for automatically controlling the flow is called a *throttling governor*.

When steam has its pressure reduced by flowing through a narrow passage, like a partly closed valve, it is often said to be *wiredrawn*.

In the second and third methods of governing, the pressure of the steam admitted to the cylinder at the beginning of each stroke is the same, and the work done in the cylinder is varied by controlling the action of the valve so as to cut off the steam supply earlier or later in the stroke. In many cases, the point

at which compression begins is also varied, which has the effect of varying the back pressure or the resistance to be overcome by the piston. If the cut-off is regulated by hand, as, for example, is the case with the link motion of a locomotive, the engine is said to have an *adjustable cut-off*. If the action of the valve is automatically controlled, the device for regulating the cut-off is called an *automatic cut-off governor*. An engine with an automatic cut-off governor is called an *automatic cut-off engine*.

5. In the steam turbine the force exerted on the rotating element can be varied only by changing the kinetic energy of the steam passing through the machine in a given unit of time. In practice the kinetic energy is changed in two different ways: First, by varying the pressure of the steam admitted to the turbine; second, by varying the weight of steam admitted without varying the pressure at which the steam passes into the turbine.

In the first method, the change in kinetic energy is effected by throttling the steam entering the turbine; this, by the lowering or raising of the initial steam pressure, changes both the velocity and weight of steam admitted in a given unit of time, or, in other words, lowers or raises the kinetic energy supplied in that time.

In practice two variations of the second method are in use. In the one variation, the steam is admitted in a continuous blast to the turbine through a number of openings; by the closing or opening of one or more of these openings by the governor, the weight of steam passing through the turbine in a given time is reduced or increased, although the steam velocity is not changed. In the second variation, the steam is admitted intermittently, that is, in puffs instead of in a continual blast. By increasing or decreasing the duration of each puff of steam in a given unit of time without changing the number of puffs, the weight of steam admitted to the turbine in that time is changed correspondingly, although the velocity of the steam is not changed. With either variation of the method under discussion the kinetic energy supplied to the turbine in a given time is changed directly in the same proportion as the weight of steam is changed.

EMERGENCY, PRESSURE, AND WATER-LEVEL GOVERNING

6. Emergency, or **safety, governors** are nearly always applied to steam turbines, and quite often to reciprocating steam engines. Such governors differ radically from speed governors, in that they have no influence on maintaining a substantially uniform speed of rotation; their purpose is to prevent automatically an excessive speed of the rotating parts.

The emergency governors used in practice may be divided into two classes. The one class consists of a supplementary speed governor operating on the same general principles as regular speed governors, but coming into action only when the speed of the rotating parts exceeds somewhat that at which the speed governor is supposed to keep it; the supplementary governor on coming into action causes a sudden and entire closing of the throttle valve, thereby stopping the engine or turbine. Safety governors of the second class, which in practice are usually applied only to reciprocating steam engines, have a device applied directly to the speed governor which comes into action automatically, if the driving connection between the speed governor and the engine breaks, and causes an entire and sudden closing of the throttle valve; emergency governors of this second class are often called *safety stops*. From the explanations given, it will be apparent that the first class of emergency governors acts after the machine attains an overspeed, while the second class acts before an overspeed is reached.

Emergency governors as constructed in practice will not open the throttle valve again after they have acted; the throttle valve must be opened by hand to restart the engine or turbine.

7. The term *engine stop* or *emergency stop* is properly applied to a device by which the throttle valve of an engine or turbine can be quickly closed from one or more distant points in case an emergency requires this to be done. Such a device in itself is not an emergency governor, but is sometimes combined with that form of a governor.

8. Pressure governors are applied to the steam end of steam pumps, air compressors, etc. in such a manner that they

operate a throttle valve in the steam supply pipe, opening this valve when the pressure in the discharge tank drops a certain amount and thereby either starting or speeding up the machine. When the pressure rises to nearly that to be maintained, the pressure governor partly or entirely closes the throttle valve, thereby slowing down or entirely stopping the machine. When pressure governors are applied to pumps, air compressors, etc. driven by an electric motor, they operate a switch that stops or starts the motor. Some pressure governors are so designed that they can only stop or start the apparatus they are applied to, and therefore they act intermittently; other pressure governors act continuously, changing the speed of the apparatus to suit variations of pressure in the discharge pipe.

From the explanations given there is seen to exist a fundamental difference between a speed governor and a pressure governor; the former is intended to maintain a uniform speed of the apparatus to which it is applied, while the latter through varying the speed of the apparatus is intended to maintain a uniform pressure in the discharge pipe.

9. Water-level governors are applied to steam pumps and also to electrically driven pumps, for the purpose of automatically maintaining a substantially uniform water level in a vessel into which they discharge. In some water-level governors, no matter whether the pump is steam-driven or electrically driven, its starting and stopping is brought about by a float that falls and rises with the water level in the vessel; in the case of steam pumps, the float through suitable means operates a throttle valve, opening this when the water becomes low and closing it when the water level has risen to the proper height, thereby starting and stopping the pump. In case of electrically driven pumps, the float operates a motor-starting switch. In other water-level governors, changes in hydrostatic pressure due to changes in water level are used to operate the throttle valve of the pump.

Both pressure governors and water-level governors when applied to pumps are often called *pump governors*.

SPEED CONTROL

CENTRIFUGAL PENDULUM GOVERNORS

10. Principle of Action.—Most speed governors depend for their action on the effect that is exerted by the centrifugal force developed in a weight revolving around an axis outside of its center of gravity. The weight is so suspended that the centrifugal force developed by its rotation is opposed by a variable resistance. When the speed and centrifugal force increase, the weight moves outwards and the resistance increases until it balances the increased centrifugal force; when the speed decreases, the centrifugal force becomes less than the resistance and the weight is forced toward the axis; as the weight moves toward the axis, the resistance decreases until it is again balanced by the centrifugal force in the new position. This motion of the weight is transmitted to a suitable mechanism which controls the flow of steam to the engine or turbine.

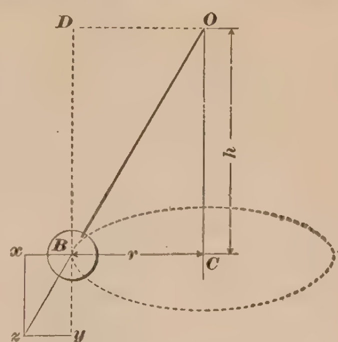


FIG. 1

11. Simple Pendulum Governor.—The earliest and at the same time one of the simplest speed governors is an application of what is known in mechanics as the revolving pendulum. It consists of a weight B, Fig. 1, suspended from the point O by a fine cord, and revolves around the vertical axis OC. When the pendulum revolves about the axis at a uniform speed, the ball remains at a constant distance r from the axis and at a constant distance OC below the point of suspension O. The latter distance is called the *height* of the pendulum and is represented in the figure by h.

When the pendulum is revolving, the ball is acted on by three forces, namely: Gravity, which is equal to the weight of the ball and acts downwards; centrifugal force, which acts

horizontally outwards; and the pull in the cord. These three forces may be respectively represented by the lines $B\,y$, $B\,x$, and $B\,z$. Considering the point of suspension O as a fulcrum, it is evident that the force represented by $B\,y$—the weight of the ball—tends to revolve the ball downwards with O as a center and a lever arm r equal to the perpendicular distance $D\,O$ from the line of action of the force to the fulcrum. Similarly, the centrifugal force—represented by the line $B\,x$—tends to revolve the ball in an upward direction around O with a lever arm equal to the height h. Since the force represented by $B\,z$ acts along a line passing through O, it has no tendency to rotate the ball around that center; it is, therefore, evident that the ball will be in equilibrium when the product of the centrifugal force multiplied by the distance h is just equal to the product of the weight of the ball multiplied by the distance r.

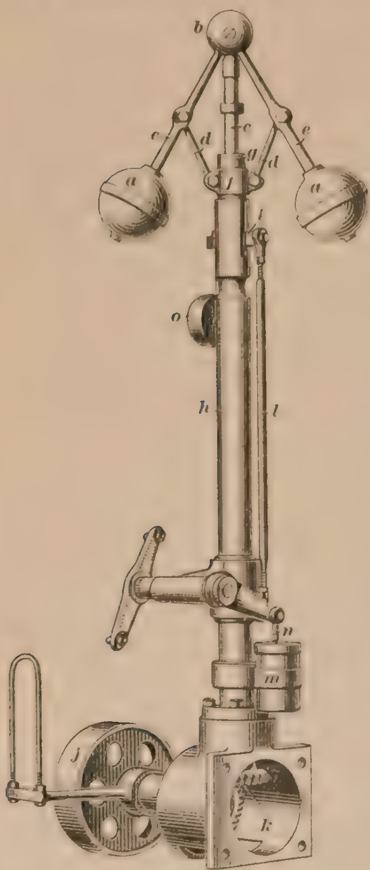


FIG. 2

If the centrifugal force is increased by an increase in the speed of rotation, the ball moves outwards; as a result, the length of the lever arm r is increased, while h is diminished until the turning effects of the two forces are again equalized.

12. Fig. 2 shows a revolving pendulum as applied to a governor used on Reynolds-Corliss engines made by the

Allis-Chalmers Company, Milwaukee, Wisconsin. For the purpose of making the governor symmetrical and preventing the bending action on the spindle that would result from the effect of a single weight, two balls *a* are used. These balls are suspended from the ball-shaped top *b*, that is fastened to and rotates with the spindle *c*. Links, or yoke arms, *d* connect the ball arms *e* to a collar, or yoke, *f* that turns freely on the sleeve *g*. This sleeve does not rotate, but is free to slide up and down the spindle *c* as well as the column *h*, being prevented from rotating by the cross-arm *i* working in vertical slots at the top of the column. In this particular governor the spindle *c*, and hence the balls *a*, are driven by a belt from a pulley on the engine crank-shaft passing over the pulley *j*, and a pair of miter gears shown at *k*. When the balls *a* fly outwards under the action of centrifugal force, the sleeve *g* is pulled upwards by the yoke *f*; this motion is transmitted to the valve gear of the engine by the side rod *l*.

The governor shown in Fig. 2 is fitted with a device called a *dashpot*, to prevent sudden fluctuations in the outward or inward motion of the balls; a dashpot is not fitted to every simple pendulum governor, however. The dashpot *m* consists chiefly of a cylinder fitted with a piston operated by a rod *n* attached to the same rocker-arm to which the side rod *l* is attached; a small opening connects the spaces above and below the piston in the cylinder, the latter being filled with water. As the piston moves up or down in the dashpot cylinder, the water passes from one side of the piston to the other, whereby the rate of motion of the piston is restrained to a low speed. Sometimes oil is used instead of water in dashpots.

The particular pendulum governor here shown is fitted with a *safety stop* contained in the case *o*; should the governor cease to revolve through an accident to its driving mechanism, the safety stop permits the balls to fall to their lowest possible position. Thereby the part of the valve gear operating the steam-admission valves is detached from the valves so that they remain closed, and thus stop the engine. While many pendulum governors are fitted with a safety stop, this practice is not universal.

13. The weights of a centrifugal governor can change their position with respect to the axis only when there is a change in their speed of rotation; consequently, such a governor cannot keep the speed of an engine uniform under all loads. For example, consider an engine with a throttling governor; when the engine is running without any load the governor valve will be opened just wide enough to admit the steam required to overcome the frictional resistances in the engine and to keep it running at a uniform speed. When the engine is loaded, the valve must be opened wider in order to admit the steam required to do the extra work. This variation in the opening of the valve can be accomplished only by a change in the position of the governor weights, and this change can take place only when the speed of the engine changes.

For some purposes, a considerable variation in speed is permissible, but in most cases it is desirable to keep the range between the speeds at no load and full load as small as is practicable; for example, in electric lighting, a total range in speed of 4 per cent. cannot be exceeded without causing a disagreeable flickering in the lights.

14. The simple pendulum governor possesses two defects that render it impracticable excepting for low-speed engines not requiring exceptionally close speed regulation. The first inherent defect is that it cannot be driven at high speed, because at high speeds of rotation its range of motion for a permissible speed variation of the engine is exceedingly small; this makes it virtually impossible to move a throttle valve or control a valve gear sufficiently to cause speed regulation. A second defect of the simple pendulum governor is that gravity, which acts in opposition to the centrifugal force developed by the rotation of the weights, is too small to quickly overcome the inertia of the weights when there is a sudden reduction in the centrifugal force. This makes the action of the governor slow in responding to a sudden increase in load; the result is that before the governor can move far enough to open the valve and admit the steam required for the greater load, the speed of the engine will drop considerably below that corresponding

to the load. Any attempt to make the governor act more promptly by increasing the weights will be futile, since the mass whose inertia must be overcome is increased in the same proportion as the increase in weight.

15. Weighted Pendulum Governor.—The simple pendulum governor has been modified in such a manner as to greatly increase the range of motion produced by the governor weights for a given range in speed; this modification is known as the *weighted pendulum governor*, and also as the *Porter governor*, from the name of its designer. The form in which this type of governor is applied to the Porter-Allen engine is shown in Fig. 3; in this governor the centrifugal weights *a* are small and their centrifugal force is made comparatively great by running them at a high speed. In addition to their own weight, the balls must lift a large weight, or counterpoise, *b* that is free to slide up and down on the spindle *c* and is lifted by means of the links *d*. The counterpoise *b* revolves with the spindle. At its lower end it is fitted with a collar that gives motion to the lever *e* that transmits the motion to the gearing that operates the valve. The lever *e* carries a weight *f* that can be adjusted along the lever so as to change the speed at which the engine will run by a small amount. The dashpot *g* checks any tendency of the governor to fluctuate too rapidly. The

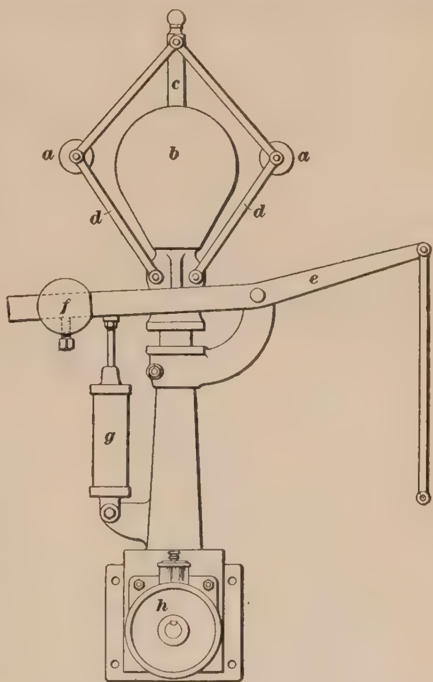


FIG. 3

governor is driven by a belt from a pulley on the engine shaft to the pulley *h*, which drives the spindle *c* by means of a pair of bevel gears.

The effect of the counterpoise is to add to the resistance, against which the centrifugal force developed by the rotation of the balls must act, without changing the centrifugal force itself, as would be the case if extra weight were added to the balls. The result is that the balls must revolve at a higher speed in order to develop a centrifugal force great enough to overcome the added resistance.

The weighted pendulum governor is much more sensitive than the simple pendulum governor, and for this reason will give a closer speed regulation.

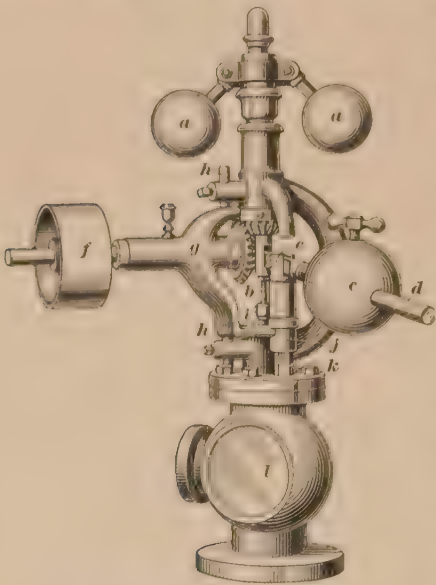


Fig. 4

16. The **Gardner Standard Class A governor** is a commercial form of a weighted pendulum governor applicable to any steam engine that is governed by throttling; it is shown in Fig. 4. The centrifugal weights *a* when moving outward under the action of centrifugal force press down the spindle *b*, to the lower end of which a double-seat poppet valve is attached, thereby partly, or even entirely, closing the steam inlet to the valve chest of the engine. Resistance to the action of the balls *a* is furnished by a weight *c* on a lever *d* fulcrumed at *e*; the one end of the lever passes through a slot in the spindle *b*. The position of the weight *c* in reference to the fulcrum can be changed, thereby changing the resistance to the action of the centrifugal weights; this changes to a large extent the average

engine speed the governor will maintain. The governor is driven by a belt from the engine passing over the pulley *f*.

A safety stop is incorporated in the construction of the governor, which automatically stops the engine if the governor belt should break or slip off its pulley. This is accomplished by mounting the horizontal driving-shaft bracket *g* on pins *h* in such a manner that it can swing on these pins; the pull of the belt holds a projection *i* of the bracket into a notch of the fulcrum sleeve *j*, which when released is free to slide on the stud *k*. Should the belt break or slip off, the projection *i* is no longer held forcibly into the notch of the fulcrum sleeve *j*, and the weight of the counterpoise *c*, lever *d*, and fulcrum sleeve *j* forces the projection *i* out of the notch; the counterpoise and fulcrum sleeve then drop and thereby close the steam inlet to the engine by forcing the double-seat poppet valve in the chamber *l* to its seat.

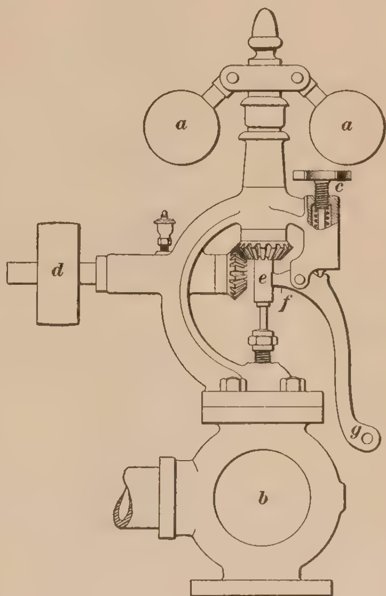


FIG. 5

17. Spring-Loaded Pendulum Governors. Springs are often used instead of a counterpoise for increasing the resistance opposing the centrifugal force of the revolving weights of a pendulum governor. Many designs of spring-loaded pendulum governors are used by engine builders; thus, in the governor shown in Fig. 2 the centrifugal weights *a* may be tied together by two helical springs, one at each side of the column, the ends of the springs being attached to the balls.

Spring-loaded throttling governors, suitable for attachment to any steam engine, are built by a number of manufacturers, each of whom employs his own designs.

18. Fig. 5 illustrates the **Gardner Standard Class B governor**, which is of the spring-loaded pendulum type and governs by throttling, the centrifugal weights or flyballs *a* operating a throttle valve in the housing *b*. The outward motion of the flyballs is resisted by a helical spring enclosed in a case provided with a hand wheel and screw *c* by means of which the amount of compression of the spring, and, in consequence, the resistance, can be varied. By changing the compression of the spring, the speed of the engine can be varied through a wide range. The spindle and flyballs are driven through bevel gears by a belt on the pulley *d*. As the balls fly outwards, they force the spindle *e* downwards; this operates the lever *f* and compresses the spring, which thus opposes the outward motion of the balls. The motion of the spindle is transmitted to the valve by means of the valve stem that is attached to the lower end of the spindle.

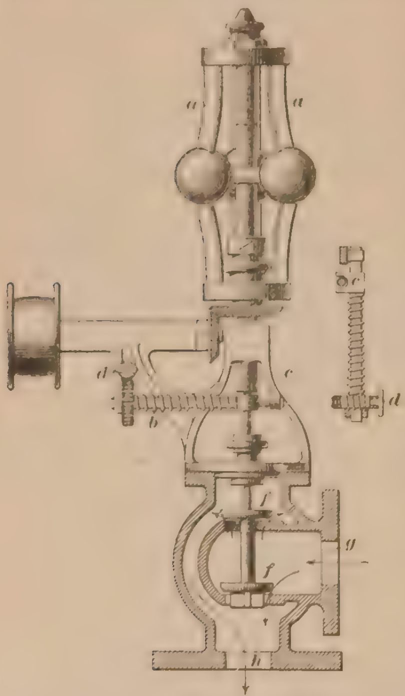


FIG. 6

The extension *g* of the lever *f* provides a means of attaching a cord or other connection by means of which the speed of the engine can be controlled from a distance. This device, which is much used in connection with engines for driving sawmills, is called a **sawyer's lever.**

19. Fig. 6 shows the **Pickering governor**, which has three flyballs attached to the flat springs *a*. In this governor the

resisting force is furnished almost entirely by the resistance of the springs and is but little affected by the weight of the balls. As a means of varying the resistance and the speed of the engine, the governor is provided with a coil spring *b* wound around a spindle that carries the fork *c*. The tension of the spring, due to having been twisted, presses this fork upwards against the governor spindle and thus increases the resistance in proportion to the spring tension, which may be regulated by the worm-wheel and worm operated by the hand wheel *d*.

As the governor balls move outwards against the resistance of the springs they lower the valves *f* and thus partly shut off the steam supply. Steam enters at *g* and flows in the direction of the arrows through the opening *h* to the steam chest. The two valves *f* are balanced; that is, the pressure on the top of one is balanced by a nearly equal pressure on the lower side of the other; this makes the resistance to their motion very small and makes it possible for the governor to move them easily.

20. The spring of the spring-loaded governor serves nearly the same purpose as the counterpoise of the weighted pendulum governor. The spring-loaded governor, therefore, shares with the weighted pendulum governor the advantages of a high speed of rotation and a high degree of sensitiveness. With the spring-loaded governor, however, the force opposing the centrifugal effect of the flyballs can be made as great as is desired without the necessity of introducing a heavy weight whose inertia must be overcome by comparatively small forces before the governor can respond to a change in speed.

Further, after the weights of a pendulum governor have been set in motion, their inertia must be overcome before they can be brought to rest in the position corresponding to the change in speed; the consequence is that if a heavy weight, like the counterpoise of a weighted pendulum governor, is used, its inertia tends to carry the governor beyond the correct position, and the valve is opened or closed more than enough to meet the change in load. This results in a change in speed that carries the governor in the opposite direction, and a series of fluctuations in speed and governor position, known as

hunting or *racing*, is set up. A dashpot tends to obviate this trouble by introducing a resistance that prevents the parts of the governor from attaining a high speed and consequently a large amount of energy to be overcome. Such a resistance, however, while it prevents violent fluctuations and hunting, makes the governor slower in adjusting itself to a change in load and so reduces the closeness of regulation that may be obtained unless a very heavy flywheel is used. With the comparatively light weights of a spring-loaded pendulum governor, the momentum of the moving parts is made small in proportion to the forces and resistances to be overcome; this type of governor, therefore, responds quickly to changes in speed and is much less subject to a tendency to hunt than the weighted pendulum governor. Another advantage of the spring-loaded pendulum governor is the facility with which the speed of the engine can be changed by changing the tension of the spring.

SINGLE-VALVE-ENGINE SHAFT GOVERNORS

21. Shifting Eccentric Along a Straight Path.—The work done in an engine cylinder can be regulated by varying either the point of cut-off or the point of compression, or by varying both of these points together. One of the most common methods of doing this in a single-valve steam engine, which uses either a plain slide valve or its equivalent, is by shifting the eccentric so as to vary simultaneously its throw and angle of advance. In practice the shifting of the eccentric is done by a governor attached to and rotating with the crank-shaft of the engine, which for this reason is called a *shaft governor*. This form of governor may depend entirely for its action on the centrifugal force of its revolving weights, in which case it is known as a *centrifugal shaft governor;* or, in addition to utilizing the centrifugal force of the revolving weights it may also employ to advantage their inertia, and then it is known as an *inertia shaft governor*.

22. To illustrate the effect on the action of a slide valve of changing the position of the eccentric with reference to the

crank and crank-shaft, consider Figs. 7 to 12. In these illustrations OC represents the crank, the motion of which is assumed to be in the direction of the arrow. The line OE represents the eccentric in the position of its greatest throw. Immediately below the circles representing the paths of crank and eccentric are shown part sections of a steam-engine cylinder with the piston P and slide valve V. In each figure the point O,

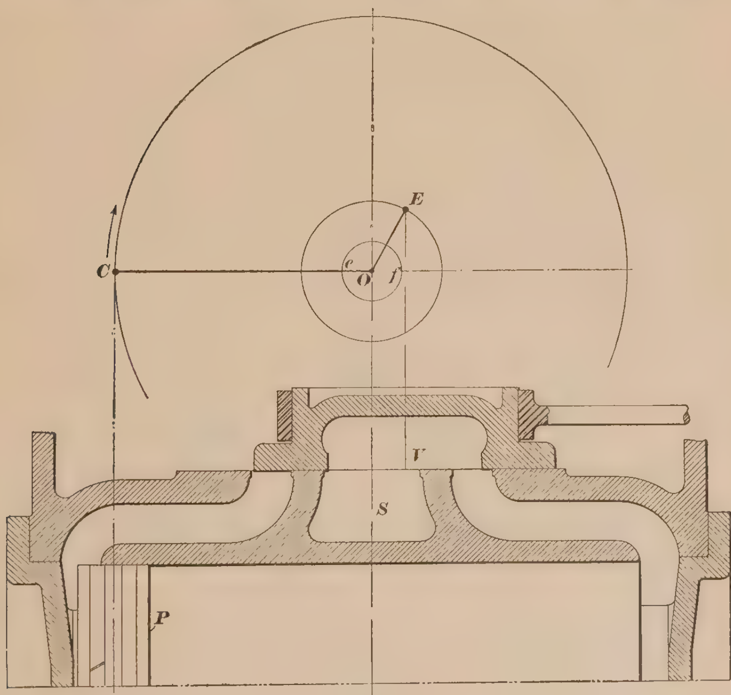


Fig. 7

representing the center of the shaft, lies in the line OS through the middle point of the valve seat, corresponding to the middle line of the valve when in its central position, and the piston and valve are in positions corresponding to those they would have if they were connected to the crank and eccentric by rods so arranged that their angularity had no effect on the relative motions of crank and piston or eccentric and valve.

In each figure the distance of the valve from its central position is equal to the perpendicular distance between the line OS and the line EV or $E'V$ drawn parallel to OS through the eccentric center positions E or E'. The radii of the small circles ef are equal to the lap of the valve; consequently, when the lines EV or $E'V$ produced are tangent to this circle, as in Figs. 11 and 12, the distance of the valve from its central

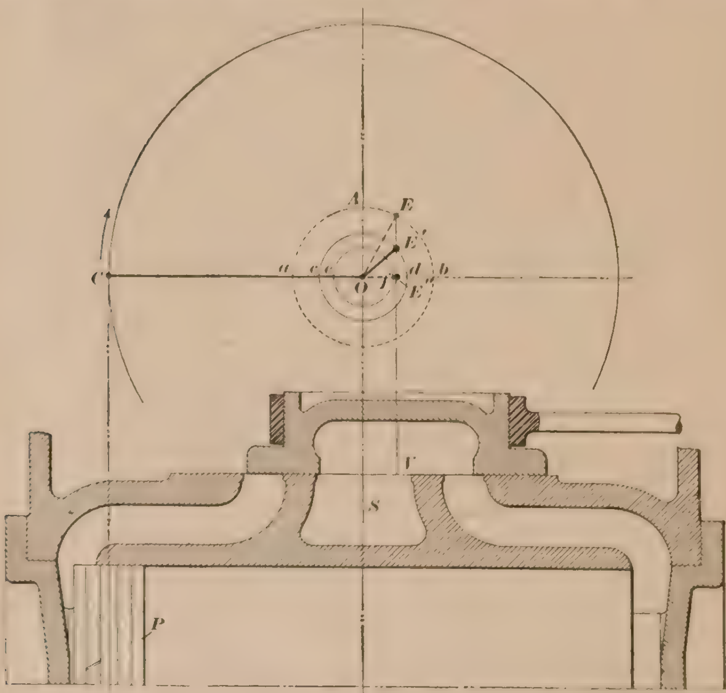


FIG. 8

position is equal to the lap, and the edge of the valve coincides with the edge of the steam port. Further, in Figs. 7, 8, 9, and 10, the distance of the line EV or $E'V$ from the extremity f of the diameter ef is equal to the port opening for the corresponding eccentric and valve positions.

23. In Fig. 8 the crank is on the center, the piston is at the left-hand end of its stroke, just ready to begin its stroke to the

right, and the valve has moved from its central position far enough to give the desired lead. Now, with the crank in the dead-center position, let the eccentric be shifted so that its center moves along a straight line at right angles to the center line $O\ C$ of the crank. An inspection of Fig. 8 shows that this movement of the eccentric does not change the position of

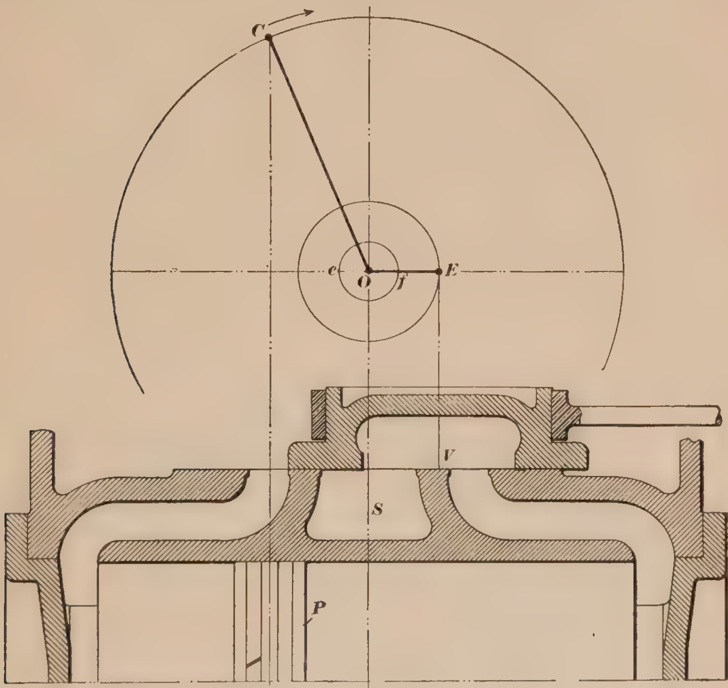


FIG. 9

the valve. With the eccentric at any point, as E' or E'', in the perpendicular through E to the center line $O\ C$, the lead of the valve is the same as when the eccentric is in the extreme outer position E; in other words, the lead of the valve is the same for all points of cut-off. When the eccentric center is in the position E'', coinciding with the line $O\ C$, the position in which the travel of the valve is least, the maximum port opening is equal to the lead. In no position of the eccentric is the valve

prevented from opening the port at the beginning of the stroke; at the position E'', however, the amount of opening is small and the period during which the valve remains open is short.

Fig. 8 shows that the change of the position of the eccentric center from E to E' has shortened the eccentric radius, thus reducing the throw of the eccentric from the distance represented by the diameter of the circle $a\,b$ to the distance repre-

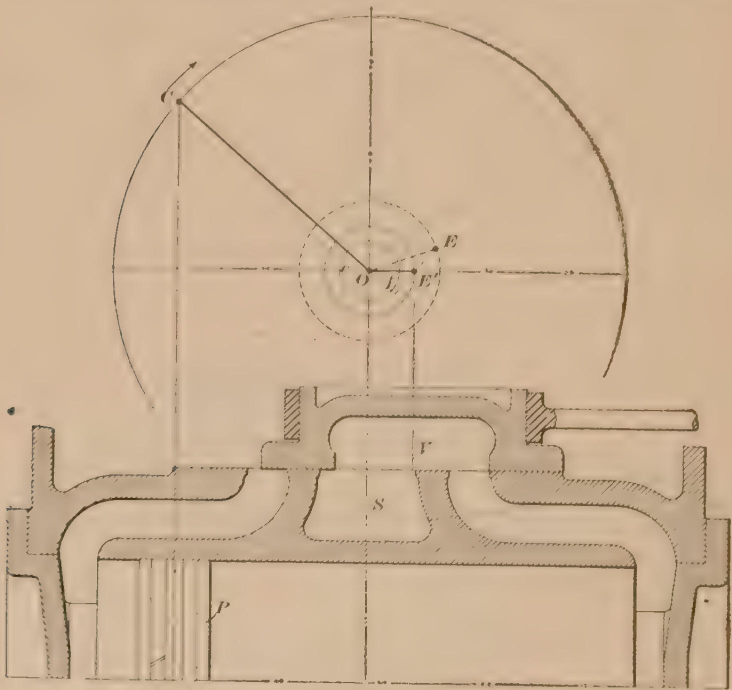


FIG. 10

sented by the diameter $c\,d$. Another effect of the change in position has been to increase the angle of advance from the value $A\,O\,E$ to $A\,O\,E'$.

24. With the eccentric in its extreme outer position, the maximum displacement of the valve is equal to the radius $O\,E$. In its extreme position, the valve opens the left port fully, as is shown in Fig. 9. Full port opening occurs when the piston

has moved through about one-quarter of its stroke from the left toward the right. The maximum displacement of the valve when the eccentric center has moved to the position E', Fig. 10, is equal to the radius $O\ E'$. In this case, the port is opened only a part of its full width. Owing to the increase in the angle of advance, the maximum port opening occurs

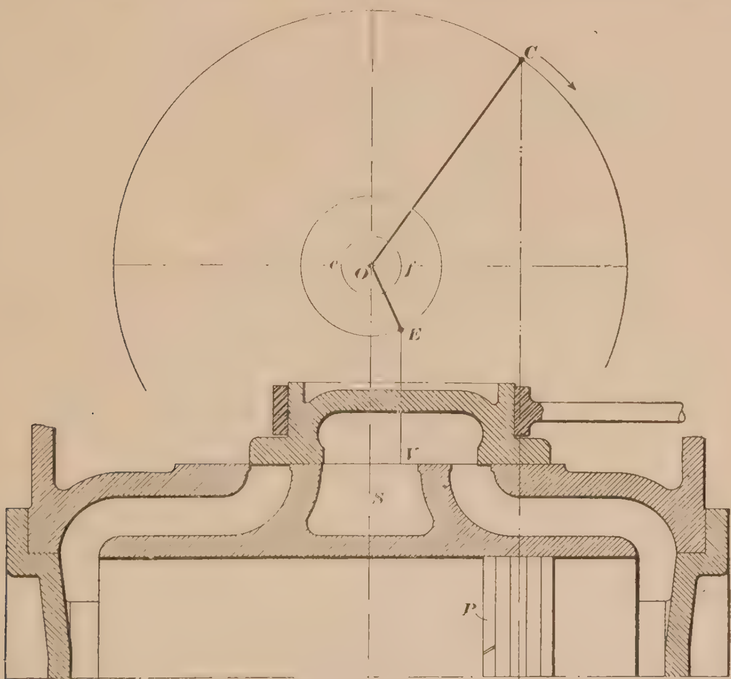


Fig. 11

when the piston has moved through only one-eighth of its stroke from the left toward the right.

25. In Figs. 11 and 12 the valve is in the cut-off position, that is, it has moved to the left until it has just closed the port to the admission of steam. In Fig. 11 the eccentric is in its extreme outer position, where it has its greatest throw, and the piston has traveled more than three-quarters of its stroke. When, however, the eccentric center has shifted to the

position E', Fig. 12, the valve closes the port and cuts off steam before the piston reaches the middle point of its stroke.

26. By a construction similar to Figs. 11 and 12, it is easily shown that the effect of shifting the eccentric from its outer position E along the line $E\ E'$ is to hasten release and compression. Thus, considering either end of the cylinder, as the

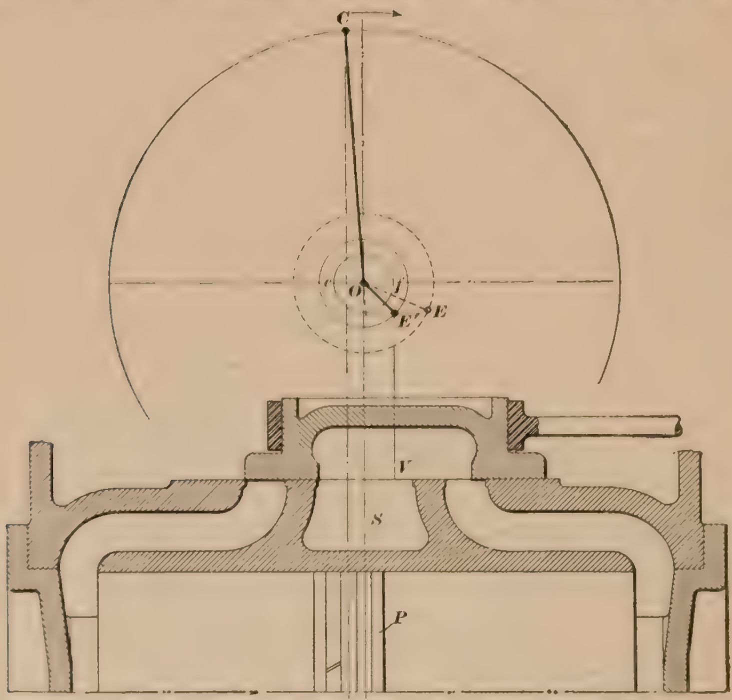


FIG. 12

eccentric center moving from E in the direction of E' approaches nearer the center line $O\ C$ of the crank, release takes place earlier in the stroke, as the piston moves from that end and the port is closed so as to prevent the escape of exhaust steam earlier in the return stroke. Compression begins earlier and the resistance to the motion of the piston is thus increased.

27. A general consideration of the effects observed in connection with the diagrams, Figs. 7 to 12, shows that the

combined effect of a reduction in the eccentric radius and an increase in the angle of advance is to reduce the valve travel and port opening and to make the events of maximum port opening, cut-off, release, and compression occur earlier in the stroke. These diagrams also show that if the eccentric is shifted in such a manner that its center moves in a line perpendicular to the center line of the crank, the lead of the valve will not be changed.

28. The centrifugal shaft governor shown in Fig. 13 is an example of shifting the eccentric in a straight line at right

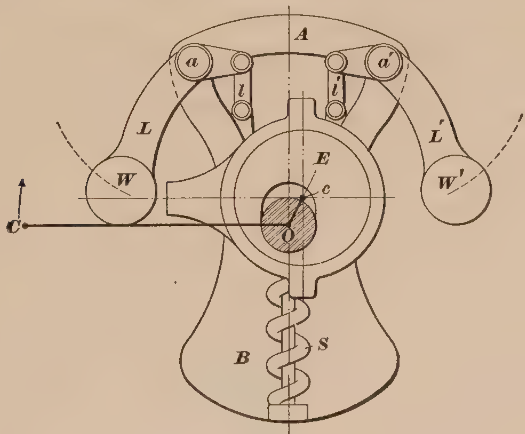


FIG. 13

angles to the center line of the crank, as was explained in connection with Figs. 7 to 12. A frame A B is keyed to the shaft O. Two bell-crank levers L, L' are pivoted to this frame at a and a'. At one end these levers are enlarged so as to form the weights W, W'; at the other end they are attached to the links l, l' that connect with the eccentric E. The eccentric is slotted, as shown, so that it may be shifted in such a manner that its center c will move in a line perpendicular to the center line O C of the crank. As the shaft with the frame and governor revolves, the centrifugal force of the weights W, W' tends to cause them to fly outwards along the dotted arcs and so force the center c of the eccentric nearer the center line O C of

the crank. This motion is resisted by the action of the spring S, which is of such a length that when put in place it is compressed and exerts a pressure that tends to hold the eccentric in its extreme outer position. This pressure, called the *initial tension* of the spring, resists the centrifugal force of the weights and prevents their moving the eccentric until the engine has reached a certain speed. As soon as this speed is exceeded, the centrifugal force of the weights exceeds the initial spring tension, and the weights move outwards, compressing the spring, until the centrifugal force and the total spring tension are equal. When the speed at which this occurs, which is the normal engine speed the governor is adjusted for, begins to be exceeded, the centrifugal force of the weights exceeds the resistance of the spring again; they then move outwards along the dotted arcs, compress the spring further, and shift the center of the eccentric nearer the line $O\,C$, thus shortening the stroke of the valve and increasing the angle of advance so as to make the cut-off, release, and compression take place earlier in the stroke. In this way the work done in the engine cylinder is reduced to such an amount that a further increase in speed is prevented. If the load on the engine increases and the speed begins to decrease, the centrifugal force of the weights is reduced until the force of the spring is sufficient to shift the eccentric in the opposite direction until enough steam is admitted to the cylinder to do the work. In this way the speed of the engine is kept nearly uniform.

29. Shifting Eccentric Along a Curved Path.—Many engineers consider it desirable to arrange the motion of the eccentric in such a way that the lead of the valve will be as near its maximum as is practicable at the cut-off position corresponding to the normal load. They also contend that the lead should be zero when the governor is in its extreme outer position and the stroke of the valve is least; that is, with the governor in this position the valve should not open the ports at all, otherwise the engine would be in danger of running away when not loaded. It has been shown that shifting the eccentric center along a straight line perpendicular to the center line of

the crank does not vary the lead. Thus, referring to Fig. 8, it is seen that the shortest throw of the eccentric occurs when the center is in the position E'' on the center line $O\,C$. In this position, the lead of the valve is the same as it was when the eccentric center was in its extreme outer position E. On account of this, it has become the common practice to guide the motion of the eccentric by attaching it to an arm pivoted

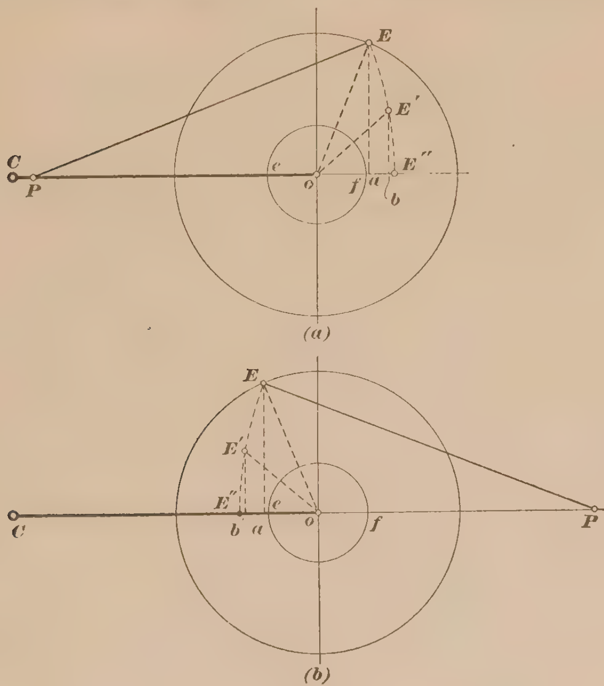


Fig. 14

either to a disk or to the arm of a wheel keyed to the shaft. The center of the eccentric then moves in an arc whose center is the center of the pin to which the arm is pivoted.

30. Figs. 14 to 16 are diagrams in which are shown the effect on the lead of varying the position of the pivot of the arm to which the eccentric is attached. In these diagrams, o represents the center of the shaft; $o\,C$ the crank; E, E', etc., the

eccentric center in various positions; and P the pivot point of the arm to which the eccentric is attached. As in Figs. 7 to 12, the radius $o f$ of the circle $e f$ is equal to the lap of the valve. In order to show more clearly the effect on the lead of a curved path for the eccentric center and a change in the point of suspension P, the eccentric radius $o E$ and the change in the eccentric position from maximum to minimum throw are shown much greater in proportion to the length $P E$ of the arm than is usual in practice.

31. In Fig. 14 is shown a method of suspension, in which the pivot P is located on the center line $o C$ of the crank. Fig. 14 (a) has been drawn for a direct valve and a direct rocker-arm, and also is correct for an indirect valve with a reversing rocker-arm. At (b), which has been drawn for an indirect valve with a direct rocker-arm, or a direct valve with a reversing rocker-arm, the pivot P of the eccentric link is on the side of the shaft opposite the crankpin. With the eccentric in its extreme outer position E the lead is least, being represented in the diagram by the distance $f a$. As the eccentric center shifts toward the earliest cut-off position E'', the lead increases quite rapidly at first, as is seen by the distance $f b$, which represents the lead when the eccentric center is in the position E'. With the pivot in this position, the lead is greatest when the eccentric center is in its point of earliest cut-off E''. The increase in lead as the cut-off becomes earlier is an advantage of this method of suspension, since at the beginning of the stroke it gives the valve a liberal opening and secures a full pressure of steam in the cylinder, thus neutralizing in some degree the tendency of the short travel of the valve to produce wire-drawing of the steam. A disadvantage is that in case the load is suddenly taken off the engine, it may attain a dangerous speed, owing to the comparatively large port opening at the position of least throw of the eccentric.

32. Fig. 15 shows a method of suspending the eccentric by which the lead is made to decrease as the eccentric center E nears the center line of the crank. The diagram (a) has been drawn for a direct valve with a direct rocker-arm or an indirect

valve with a reversing rocker-arm, while diagram (*b*) has been drawn for an indirect valve with a direct rocker-arm or a direct valve with a reversing rocker-arm. This method of suspending the eccentric, however, has the disadvantage of wiredrawing the steam at early cut-offs. With this disadvantage, however, is combined the advantage that in case all load should suddenly be thrown off the engine, the restricted port opening due to

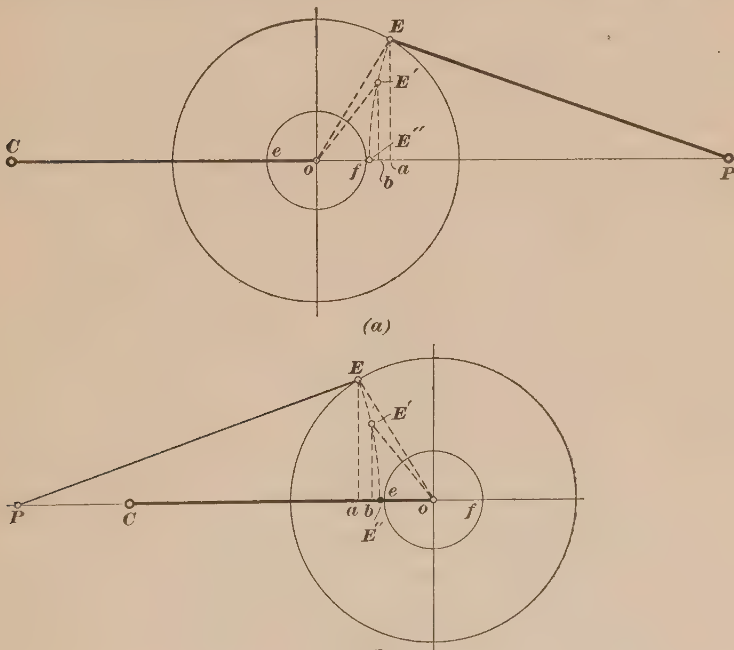


FIG. 15

the eccentric being shifted to its position of least throw will tend to prevent the engine from attaining a dangerous speed, and this fact has served to make this manner of suspension a favorite with builders of single-valve engines.

33. Fig. 16 shows a method of suspending the eccentric that combines the advantages of the two methods of suspension that were shown in Figs. 14 and 15. In this method the pivot of the eccentric link is not located on the center line of the

crank but to one side of it. As in Figs. 14 and 15, the diagram (*a*) has been drawn for a direct valve with a direct rocker-arm or an indirect valve with a reversing rocker-arm, while (*b*) has been drawn for an indirect valve with a direct rocker-arm or a direct valve with a reversing rocker-arm. With this method of suspension the change in lead is very slow during the motion of the eccentric center through the range corre-

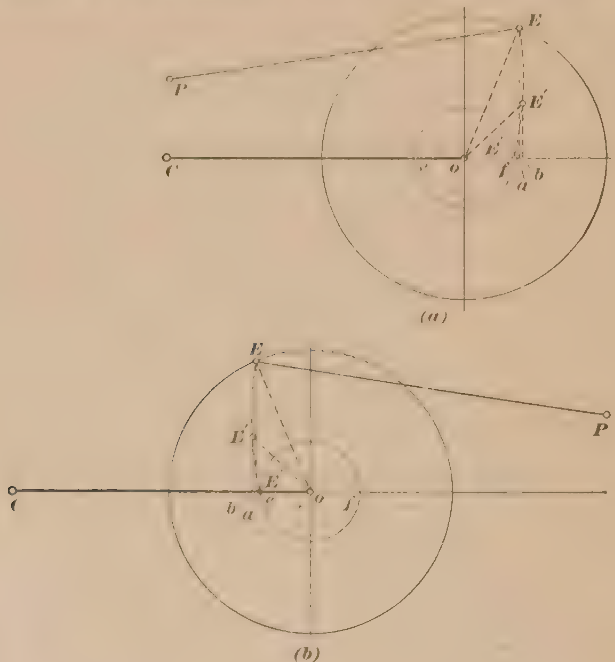


FIG. 16

sponding to the usual load. When, however, the eccentric center approaches the position of minimum throw, the lead is rapidly reduced and can readily be made zero at the limit.

34. The centrifugal shaft governor used on some Westinghouse engines, which is shown in Figs. 17 and 18, embodies the method of suspension of the eccentric that was shown diagrammatically in Fig. 14 (*b*), a direct-connected indirect valve being used on this engine. A disk *A* is attached to the shaft and the

weights B and B' are pivoted to this disk at b and b'. The eccentric E is attached rigidly to the arm c, which is pivoted to the disk at d. The two weights are connected by a long link S in such a way that they always move together. The short link f connects the weight B with the eccentric; any motion of the weights is thus transmitted to the eccentric so as to change the position of its center with respect to the center line $O\ C$ of the crank.

Fig. 17 shows the weights in their inner position, where the eccentric is in the position of its greatest throw. As the shaft

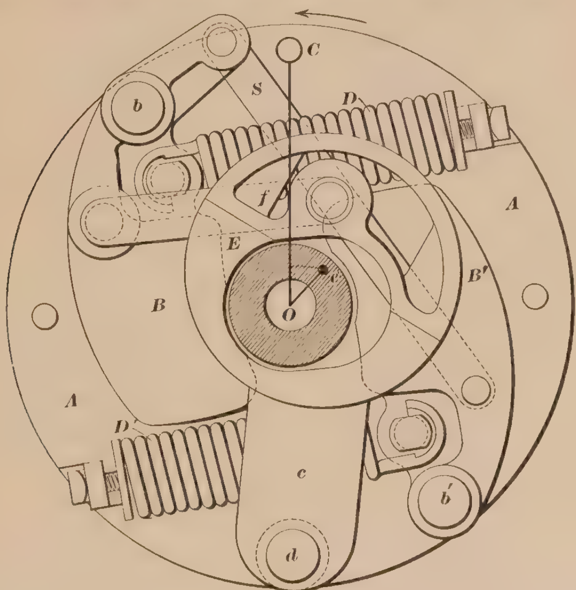


FIG. 17

revolves, the centrifugal force of the weights is resisted by the tension of the springs D, D'. When the speed becomes great enough to enable the centrifugal force to overcome the resistance of the springs, the weights move outwards and shift the eccentric about its pivot d until the travel of the valve is adjusted to the work to be done. Fig. 18 shows the governor when the weights are in their extreme outer position. The eccentric

center then takes the position e', nearly on the center line $O\ C$ of the crank, having moved from the position e of Fig. 17 along the dotted arc; this is the position of earliest cut-off.

35. Effects of Inertia on Centrifugal Governor Action.—In centrifugal shaft governors the force that acts to shift the valve or eccentric is principally the difference between the centrifugal force of the revolving weights and the resistance opposed to this force by gravity or the governor

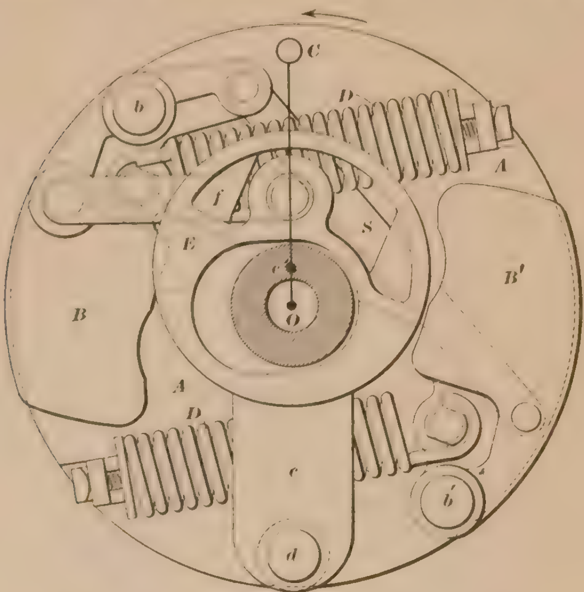


FIG. 18

springs. Inertia of the moving parts of the governor, however, has an influence on the changing of their position, the effect of inertia being first to oppose the motion of the governor weights, and then, after the weights have been set in motion, to carry them beyond the point at which there is equilibrium between their centrifugal force and the resistance. Inertia, thus, has the effect of a disturbing force; it makes the governor slow to respond to the action of the forces induced by a change in the speed of rotation, and thus permits a considerable change in

speed before the governor adjusts itself to the change in load; it also acts to carry the weights beyond the point corresponding to the correct valve opening or eccentric position, and so sets up a series of vibrations and the governor hunts or races.

In the spring-loaded pendulum governor, inertia is made small by the use of small weights, while the useful forces are made comparatively large by running the governor at a high speed and using powerful springs; in this way the disturbing effect of inertia is kept from becoming serious. In the shaft governor, however, the speed of rotation is limited to the speed at which the engine may run; it is, therefore, impossible to obtain enough centrifugal force to control the valve properly without the use of heavy weights, and this, in turn, means a corresponding increase in inertia.

36. Construction of Inertia Governors.—It has been found possible to so construct a governor that the effect of inertia instead of acting as a disturbing force is made to assist the weights in adjusting themselves to changes in speed. Such a governor can, therefore, be made to respond very promptly to the slightest variation in engine speed and at the same time be free from a tendency to hunt; it is known as an *inertia governor*. This form of governor is very widely used today as a shaft governor on high-speed steam engines, where very close speed regulation is desired; in this connection, it naturally revolves at crank-shaft speed. Inertia governors greatly resembling the shaft governor, but running in a horizontal plane, are used for steam-turbine regulation as well; when thus applied, the governor runs very much slower than the turbine rotor, however, and usually operates by throttling the steam supply.

37. The basic principle of the inertia governor is shown in Fig. 19; this illustration, it must be clearly understood, does not show a governor as actually constructed, but merely serves to point out how the effect of inertia can aid in governing. A weight arm, consisting of a bar *a* carrying two weights *b*, is pivoted at *c* to an arm of the flywheel. The point *c* in this case is also the center of gravity of the weight arm, and this point

always remains at the distance $d\,c$ from the shaft center d. If the wheel turns at a uniform speed in the direction of the arrow e, there will be no change of position of the weight arm relative to the wheel, and all parts will turn about d at the same rotative speed. If a sudden increase of load should cause the engine to slow down, the weight arm, by reason of its inertia, will turn about the pivot c in the direction of the arrows f; for the weights b, having the same rotative speed as the wheel before the change of load occurred, will tend to keep on revolving at that speed, which will result in swinging the weight arm for-

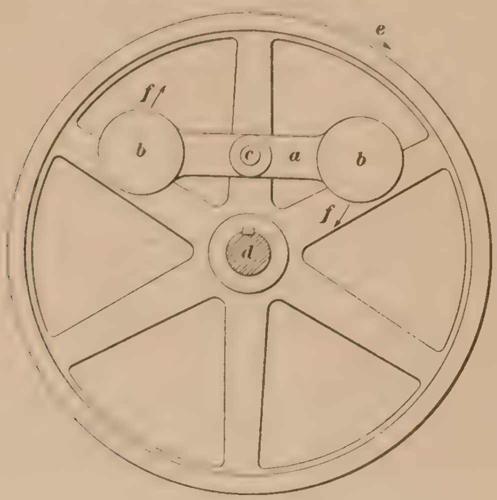


Fig. 19

wards on the pivot c. This motion of the weight arm relative to the wheel is utilized to alter the point of cut-off in the steam engine. Should the speed increase, the weight arm will lag behind and will turn in the opposite direction about the pivot c.

38. There are many forms of inertia shaft governors used in practice; in all of them both the centrifugal force as well as the inertia of a revolving weight is utilized.

Inertia shaft governors are broadly divided into two general classes, which are the weighted-bar and the weighted-leaf-spring classes; the first class is known as *Rites inertia governors*,

and the second as *Armstrong inertia governors*, from the names of their original designers. Inertia governors are also classed as *balanced* and *unbalanced* governors; a balanced inertia shaft governor is so constructed that it is not affected by gravity, while the unbalanced governor is affected by it. If an inertia shaft governor is unbalanced it will not be sensibly affected by gravity when running at speeds in excess of 250 revolutions per minute, but the weight bar or weight is liable to oscillate at low speeds, such as occur in starting and stopping an engine.

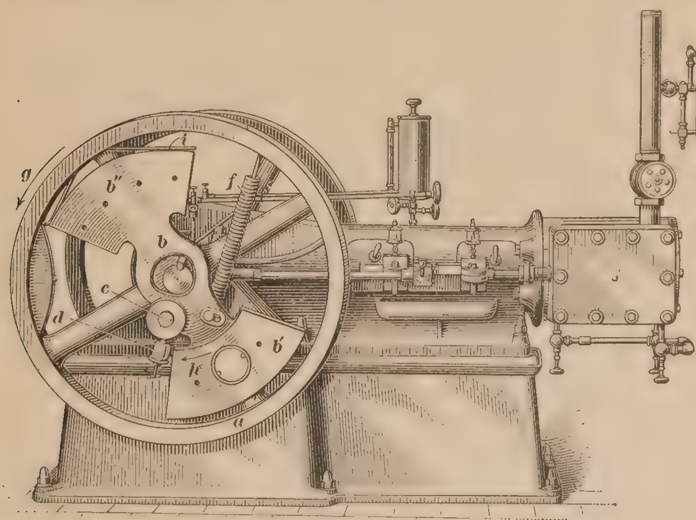


FIG. 20

When the normal engine speed is less than 250 revolutions per minute, it is the common practice to fit a balanced inertia shaft governor; when the normal engine speed is higher, practice varies, some manufacturers fitting a balanced and some an unbalanced governor.

39. A **Rites** unbalanced inertia shaft governor, as applied to high-speed steam engines built by the Troy Engine and Machine Company, Troy, Pennsylvania, is shown in Figs. 20 and 21; in Fig. 20 a side view of the engine is presented and in Fig. 21 the wheel carrying the governor has been removed from

the crank-shaft and turned around in order to show the manner in which the eccentric is suspended. The same parts have been given the same reference letters in both illustrations, as far as possible, and both should be referred to in connection with the description. The belt pulley a, which also acts as one of the two flywheels, and which is sometimes called the *governor case*, is keyed to the crank-shaft. The inertia bar b is pivoted by the shaft c to the governor case; the shaft c is free to turn in a bearing lubricated by the grease cup d, and carries on the inside an arm on which the eccentric sheave e is formed. As both the inertia bar b and the eccentric sheave e are fastened to the shaft c, any rotation of the inertia bar causes the eccentric to move in a curved path. The end b' of the inertia bar is somewhat heavier than the end b''; a helical spring f is attached to the rim of the governor case a, as well as to the inertia bar, in such a manner that its tension can be changed. As the governor case revolves as shown by the arrow g, the speed of the engine increases until the centrifugal force of the inertia bar and the tension of the spring f are equal. When the speed exceeds that determined by the initial tension of the spring f, the weighted end b' of the inertia bar rotates around its pivot at c in the direction of the arrow h, thereby shifting the eccentric center nearer the center of the crank-shaft and hence causing cut-off to occur earlier in the stroke, thus decreasing the engine speed. Up to this point the action of the inertia shaft governor is the same as that of a centrifugal shaft governor.

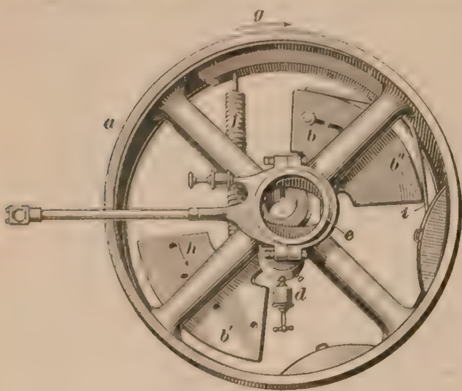


FIG. 21

Now suppose that there has been a sudden large decrease in the load on the engine, which would cause a sudden speed

increase if no governor was fitted. Then, as the load, and consequently the resistance to the turning of the shaft, suddenly drops, the speed of rotation of the shaft and governor wheel quickly begins to increase. The only way in which this increase in the speed of rotation can be imparted to the inertia bar *b* is through the action of the flexible spring *f*. Owing to the weight and length of the bar, its inertia offers considerable resistance to a sudden change in its angular velocity; the result is that the wheel advances faster than the bar, which has the effect of shifting the eccentric *e* nearer the center of the shaft and thereby causes the cut-off to take place earlier, thus preventing a serious increase in speed before the centrifugal force can act. If the load on the engine is suddenly increased, the speed of the wheel is quickly checked; the resistance of the bar to a sudden change in its speed of rotation causes it to move ahead of the wheel and thus shift the eccentric so as to make the cut-off take place later; in this way enough steam is admitted to the cylinder to meet the demand. It is thus seen that in case of a change in the speed of the shaft, the inertia of the bar acts at once to shift the eccentric in the required direction, and so prevents the change from becoming serious before the centrifugal force can

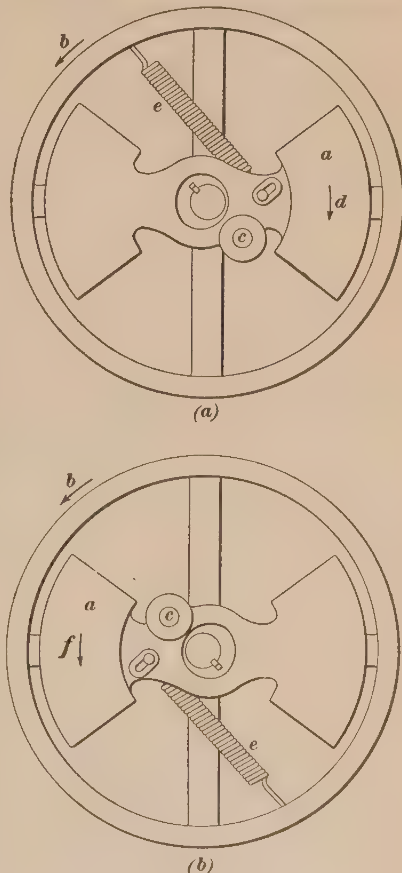


FIG. 22

operate. As soon as the eccentric reaches the position corresponding to the work to be done, the speed of the engine is controlled, as before, by the relation between the centrifugal force of the weighted end b' and the tension of the spring f.

The suspension of the eccentric in Fig. 20 is that shown in Fig. 15 (*a*) for a direct valve; consequently, the lead decreases as the cut-off becomes earlier.

40. The Rites inertia shaft governor shown in Fig. 20 is not balanced against gravity; the manner in which gravity

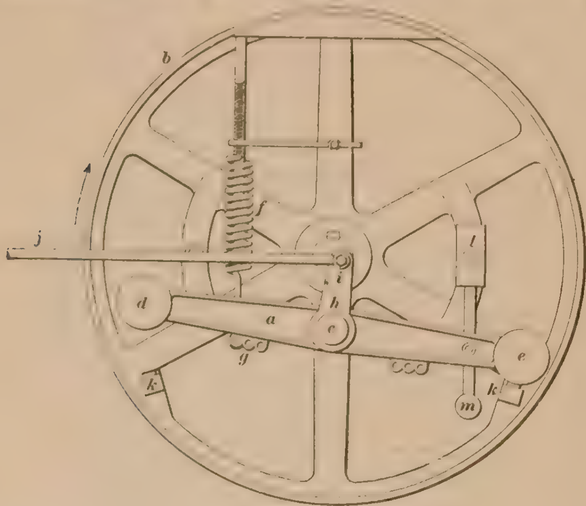


Fig. 23

affects an unbalanced inertia shaft governor rotating in a vertical plane is shown in Fig. 22. The end *a* of the inertia bar being the weighted end, when the governor case is in the position shown in (*a*) and turns in the direction of the arrow *b*, gravity tends to pull the inertia bar around its pivot *c* in the direction of the arrow *d*, thereby increasing the pull on the spring *e*. When the governor case has turned to the position shown in (*b*), gravity tends to pull the inertia bar around its pivot *c* in the direction of the arrow *f*, thereby lessening the pull on the spring *e*. Consequently if no means are provided for preventing this, the inertia bar oscillates, that is, rocks back and

forth, on its pivot with every revolution of the crank-shaft. The oscillation will be very marked at very low speeds, and may be large enough to swing the inertia bar through its whole range of motion.

To prevent, or at least to dampen, oscillation of the inertia bar of an unbalanced inertia shaft governor, some form of brake is employed by many engine builders; in the Rites governor shown in Fig. 20, a spring *i* rubbing against the end b'' of the inertia bar is used for this purpose.

If an inertia speed governor runs in a horizontal plane, as is the case with some steam-turbine governors, it needs no balancing against gravity, as it is not affected by it.

41. The **Begtrup governor** used on some McEwen steam engines, built by the Ridgway Dynamo and Engine Company, Ridgway, Pennsylvania, is a modification of the Rites inertia governor, but operates on the same principle; it is shown in Fig. 23 in the position occupied when the engine is standing still. The inertia bar *a* is pivoted to the governor case *b* at *c*; the end *d* of the inertia bar is heavier than the end *e*. The spring *f*, the tension of which can be changed, resists the centrifugal force developed by the weight *d* when the engine is running; this spring is attached at one end to the governor case and at the other end to the inertia bar at *g*. As is the usual practice with inertia governors, the point of attachment of the spring to the inertia bar can be changed in relation to the center of rotation of the inertia bar, thereby changing the length of the lever arm at which the pull of the spring acts; this is only made use of in adjusting the governor action. The inertia bar *a* carries a crank-arm *h* to which the pin *i* is attached; this pin, known as the *eccentric pin*, serves the same purpose as an eccentric, being its mechanical equivalent, and drives the valve through the valve rod *j*. The eccentric-pin suspension is that shown in Fig. 15, whereby the lead is decreased as the cut-off becomes earlier.

The two stops *k* limit the motion of the inertia bar. The governor not being balanced, a brake in the form of a dash-pot *l* is employed to check oscillation of the inertia bar; the

dashpot consists of a cylinder in which works a loosely fitting piston, which connects to the inertia bar. A small weight *m* counterbalances the centrifugal force of the dashpot *l* and its piston.

42. The general principle involved in constructing a balanced Rites inertia governor is clearly exhibited in the **Fleming governor** used on steam engines built by the Harrisburg Foundry and Machine Works, Harrisburg, Pennsylvania; this governor is shown in Fig. 24 in the position occupied by the parts when the engine is not running. The governor case *a* carries two inertia bars *b* symmetrically disposed in it and pivoted at *c*; the two inertia bars are linked together and also to the eccentric arm *d*, pivoted at *e*, by the links *f*. Each inertia bar has its own tension spring *g*, the tension of which is adjustable; the point of attachment of the springs *g* to the governor case is also adjustable, whereby the length of the lever arm at which the pull of the spring acts can be varied in adjusting the governor. The weighted end of each inertia bar is at *i*, directly opposite each other, and consequently the action of gravity on one weight is balanced by its action on the other weight for all positions of the governor. The eccentric suspension is that shown in Fig. 15, where the lead decreases as the cut-off decreases.

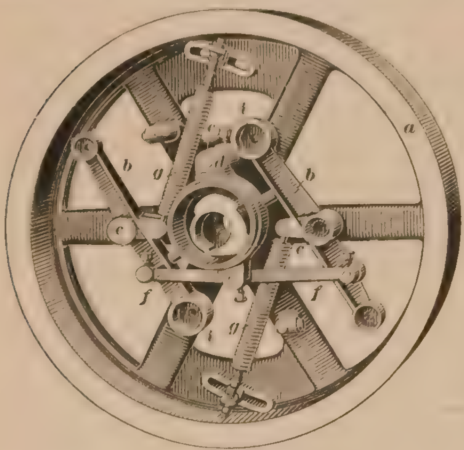


FIG. 24

43. An **Armstrong** inertia shaft governor used on some engines built by the Ball Engine Company, Erie, Pennsylvania, is shown in Fig. 25. This governor is unbalanced, and is characterized by the absence of the inertia bar of the Rites governor;

the governing element consists of a weight *a* carried on the end of a leaf spring *b* one end of which is rigidly fastened to the governor case *c*. The weight *a* is connected by a link *d* to the eccentric arm *e* pivoted to the governor case at *f*. The eccentric arm carries an eccentric sheave shown at *g*; the method of suspension is that shown in Fig. 15, and consequently the lead decreases as the governor makes cut-off earlier.

The action of the Armstrong governor does not differ from that of the Rites governor, excepting that the temporary

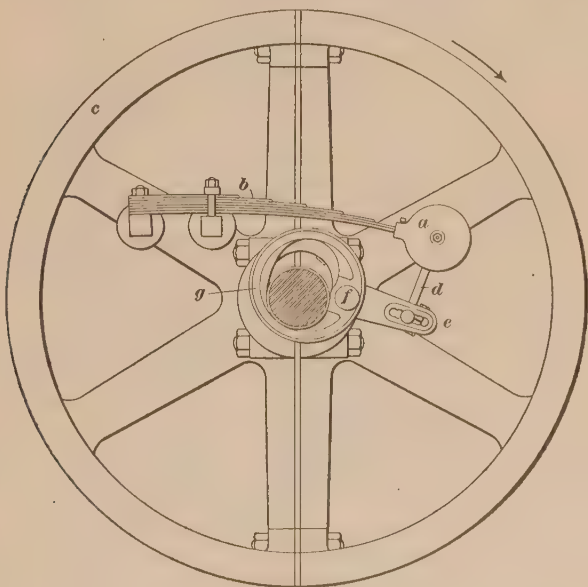


FIG. 25

lagging behind of the weight *a* under a speed increase and running ahead under a speed decrease takes place through bending of the spring *b*.

44. A balanced Armstrong inertia governor is used on some engines built by the Erie City Iron Works, Erie, Pennsylvania. This governor, which is shown in Fig. 26, is in reality a combination of the Armstrong and the Rites governors, because it combines the weighted leaf spring *a* of the one type with the

inertia bar *b* of the other type. The effect of gravity on the weight *c* is balanced by its effect on the weight *d* on the end of the inertia bar *b*. The eccentric pin is carried on an arm of the inertia bar, and the weight of this pin and arm is balanced by the weight *e*. The eccentric pin is suspended as shown in Fig. 15, and consequently the lead decreases as the governor makes the cut-off earlier.

45. If an eccentric suspended as shown in Fig. 14 is to be used, in order to have the lead increase as the cut-off is made

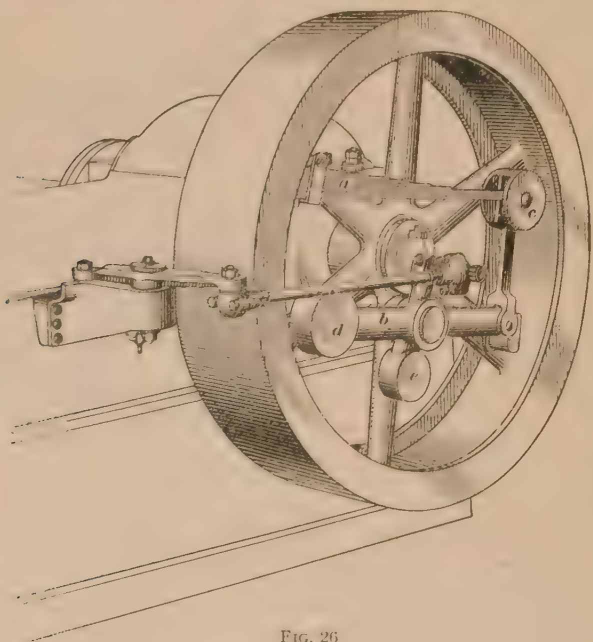


FIG. 26

earlier by the governor, the eccentric sheave or eccentric pin must be carried by an eccentric arm separate from the inertia bar of the Rites governor and so connected to that bar that its direction of rotation, with respect to the governor case, will be opposite to that of the inertia bar. In the Armstrong inertia governor the motion of the governor weight must be opposite in direction to that of the eccentric arm in order to permit an

eccentric suspension giving an increasing lead as the cut-off is shortened.

Fig. 27 shows how this principle is applied to the Rites inertia governor of some steam engines built by the American

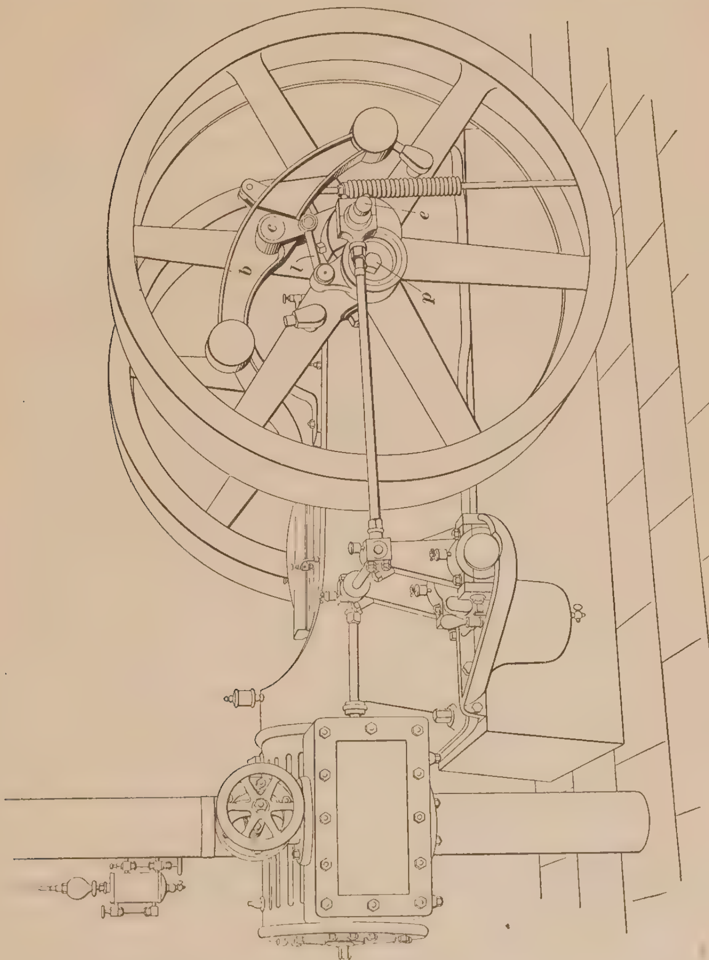


Fig. 27

Engine Company, Bound Brook, New Jersey. The eccentric pin *e* is attached to an arm that is pivoted at *p*. The inertia bar *b* is pivoted at *c* and is connected to the eccentric arm by the

link l. The engine shown runs over; that is, when viewed from the right-hand side as shown, the flywheels turn clockwise. An inspection of the figure shows that the direction of rotation of the eccentric arm around the pivot p is opposite that of the inertia bar around its pivot c.

MULTIPLE-VALVE-ENGINE SHAFT GOVERNORS

46. Steam engines of the two-valve type usually have one main valve which controls admission and exhaust of the steam, and a separate cut-off valve operating on the main valve which only determines the point of cut-off. Engines of the four-valve type may have four poppet valves, four rotary valves, or four gridiron valves with separate cut-off valves riding on top of the two steam valves. In practice, multiple-valve engines, that is, engines employing more than one valve for the distribution of the steam, are governed by pendulum governors and also by shaft governors; the latter may be purely centrifugal governors, or may be of the inertia type. Many shaft governors applied to multiple-valve engines operate on a separate eccentric actuating only the cut-off mechanism, while others operate on an eccentric actuating all valves. The eccentric may be suspended so as to move in a straight path or in a curved path, both its throw and its angle of advance being changed by the governor, just as in single-valve engines, and the same forms of governors employed for these engines may be used. Some multiple-valve engines, however, have the valve gear so arranged that only the point of cut-off is changed by the governor, the other events in the steam distribution remaining constant; with such engines, changes in the cut-off may be effected by merely changing the angle of advance of the cut-off eccentric without changing its throw, or, in other words, by rotating the eccentric around the crank-shaft.

47. An example of a shaft governor changing the angle of advance is found in the Buckeye engine built by the Buckeye Engine Company, Salem, Ohio. This engine is of the two-valve type employing a piston main valve with a riding cut-off

valve inside the main valve; the cut-off valve is actuated by a separate eccentric controlled by the governor.

The governor itself is shown in Fig. 28, with its parts in the position occupied when the engine is at rest. The two governor weights *a* are mounted on the arms *b* pivoted at *c* to the governor case *d*; the weights can be moved along the arms *b* for adjusting the governor and then locked in position. The centrifugal force

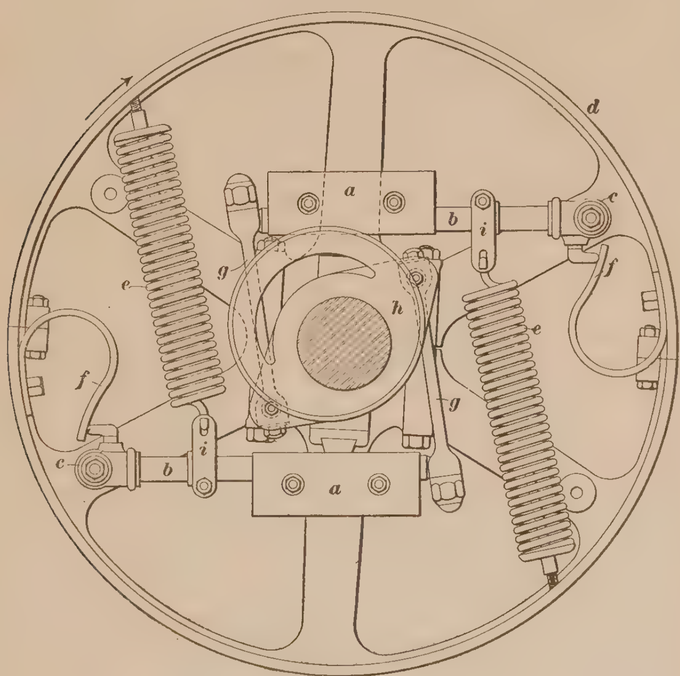


Fig. 28

of the weights is resisted by the two tension springs *e*; one end of these is fastened to the governor case *d* and the other end to the arms *b* in such a manner that the point on the lever arm at which the pull of the springs is exerted can be changed in adjusting the governor. Two auxiliary springs *f* act in opposition to the main springs, thereby lessening their tension, until the weights have moved outwards through the first half of

their range; during the outer half of the range of motion of the weights the auxiliary springs cease to act and the weights are affected only by the tension of the main springs. The auxiliary springs acting in opposition to the main springs with a force that decreases from the beginning of the outward motion of the weights until mid-position is reached, make it possible to give the main springs a higher initial tension. This secures a greater degree of sensitiveness during the outer half of the range of motion of the weights than could be obtained without causing unsteadiness and hunting during the inner half of the range. The two weight arms *b* are attached by links *g* to the eccentric *h*, which is free to turn on the crank-shaft, and consequently any inward or outward motion of the weights *a* causes a turning of the eccentric.

ADJUSTMENT OF SPEED GOVERNORS

48. Adjustment of Simple Pendulum Governors. The range of speed through which a simple pendulum governor will have a satisfactory control of the valve or valves of an engine is very limited, and no change in the weight of the fly-balls or in their position on their arms has any effect on the position in which it holds the valve or valves for a given speed. Owing to this fact the only method of changing the speed of an engine controlled by a simple pendulum governor, that is, the average speed the governor permits, is to change the ratio between the diameters of the governor pulley and the pulley on the engine shaft. If the speed at which the engine runs and the diameter of the governor pulley are known, the diameter of the governor pulley for another speed of the engine may be calculated by the formula

$$a = \frac{b\,c}{d}$$

in which a = diameter of new governor pulley;
b = diameter of old governor pulley;
c = desired speed of engines;
d = old speed of engine.

EXAMPLE.—With a governor pulley 6 inches in diameter, an engine runs at a speed of 175 revolutions per minute; what must be the diameter of pulley if the engine is to run at a speed of 190 revolutions per minute?

SOLUTION.—Applying the formula,

$$a=\frac{6\times 190}{175}=6.514\text{ in.}=6\tfrac{33}{64}\text{ in., nearly. Ans.}$$

49. Adjustment of Weighted Pendulum Governors. Speed governors of the weighted pendulum type are designed to run at a certain speed in order to maintain a given normal engine speed and will work best when run at that speed. However, if they are provided with an adjustable weight as shown at f in Fig. 3 and c in Fig. 4, the average engine speed can be changed to some extent, either by shifting the weight on the lever, or by changing its weight. Shifting the weight nearer the end of the lever or increasing its weight adds to the resistance opposing the centrifugal force of the flyballs, and thereby permits the engine to run at a higher average speed; conversely, shifting the weight in an opposite direction or making it lighter causes the engine to run at a lower average speed.

If it is desired to make a large change in the average speed of an engine controlled by a weighted pendulum governor, it is advisable to change the size of the governor pulley in accordance with the formula in Art. 48.

50. Adjustment of Spring-Loaded Pendulum Governors.—On account of the facility with which the resistance to centrifugal force can be varied by varying the tension of the spring, a considerable change in the average engine speed may be obtained with most spring-loaded pendulum governors without the necessity of changing the size of the governor pulley. Changes in the tension of the spring, however, have an effect on the sensitiveness of the governor. With a certain tension and speed, the relation between the centrifugal force of the weights and the resistance offered by the spring will give the most satisfactory results obtainable, and any variation from this will tend either to reduce the sensitiveness or to make the sensitiveness so great that the governor becomes unstable. For this reason, it is always desirable to run the governor as nearly the speed for which it was designed as is practicable and

to use a combination of pulleys that will secure this speed. If it is desired to make a considerable permanent change in the speed of the engine, such change should be made by changing the size of governor pulley in accordance with the formula in Art. **48**, and not by changing the tension of the spring.

51. Adjustment of Shaft Governors.—The average engine speed that will be maintained by a shaft governor of the centrifugal as well as of the inertia type is that at which the centrifugal force developed by the weight or weights of the governor exactly balances the resistance of the spring or springs opposing the outward motion of the weights. From this it follows that, the spring resistance not being changed, reducing the weight or weights increases the engine speed, and increasing the weight or weights decreases the engine speed. If the governor weights are adjustable in reference to their pivots, as for instance in the Buckeye governor shown in Fig. 28, the centrifugal force of the weights is not changed by shifting them on their arms, but the effect of the centrifugal force on the spring is materially changed. Thus, shifting the weights toward their pivots decreases the effect of their centrifugal force on the springs and therefore increases the average engine speed; conversely, shifting the weights farther out from their pivots decreases the average engine speed.

It will be obvious that a change in the spring resistance to the centrifugal force of the weight, that is, a change in the strength of the spring, will also produce a change in engine speed; making the spring stiffer or increasing its effect in opposing the weights causes the speed to increase; and making the spring weaker or decreasing its effect in opposing the weights decreases the speed at which the governor will hold the engine.

The sensitiveness of a shaft governor is determined by the initial tension of the spring or springs opposing the action of the governor weights; if the governor is slow in acting, an increase in the initial tension will make it act quicker, that is, make it more sensitive; and a decrease in the initial spring tension will cause the governor to act slower if it is found to be too sensitive, so as to cause hunting.

The resistance that a spring offers to the action of the governor weight or weights depends on two factors; these are its deflection, that is, the amount it is elongated or compressed after the initial tension has been overcome, and the initial tension. Therefore, a change in the initial tension also changes the spring resistance; but, as previously stated, a change in the initial tension also affects the sensitiveness of the governor. Hence it is inadvisable to try to change the engine speed by changing only the initial tension of the governor spring or springs, as by doing so the governor may become so sensitive or so sluggish as to become worthless. If the governor is neither too sensitive nor too sluggish, a very small change of average engine speed, say not more than 2 per cent. from the previous speed, can usually be effected by a change in the initial spring tension, but for any larger speed change either the weights or their effect, or the stiffness of the springs or their deflection, should be changed.

52. In most shaft governors, provision is made for a ready change of the governor weight or weights; thus, in many cases the governor weight or weights are made hollow and partly filled with shot, while in others flat disks may be removed or added. In still others the weights are movable in respect to their pivots, so that the effect of their centrifugal force on the spring resistance can be increased or decreased. In practically all shaft governors using helical springs, a screw adjustment, readily discernible by inspection, is provided for adjusting the initial tension of the spring. Many centrifugal governors, and practically all Rites inertia governors, are also provided with means of changing the deflection of the spring, which is the equivalent of substituting a stronger or weaker spring. Thus, in the Buckeye governor shown in Fig. 28, the one end of each spring is attached to a collar *i* on the arm *b*, which collar is movable along the arm *b* and can be clamped thereto. Shifting the collars *i* toward the pivots *c* is equivalent to fitting weaker springs, while shifting the collars away from the pivots is equivalent to fitting stronger springs. In Rites inertia governors either the end of the spring attached to the inertia bar can be shifted in reference to the pivot of the inertia bar, as clearly

shown in Figs. 20, 22, and 23, or the end attached to the governor case can be shifted in a slot, as shown in Fig. 24, so as to change the length of the lever arm at which the spring acts on the inertia bar. Inertia governors of the Armstrong type are seldom provided with any ready means of adjustment; changes in initial spring tension usually have to be made by inserting or taking out shims from between the leaf spring and one of its seats. In some Armstrong inertia governors, as for instance in that shown in Fig. 25, the end of the link joining the governor weight to the eccentric arm can be shifted in reference to the eccentric arm fulcrum, thus changing the amount of movement of the eccentric center across the shaft in reference to the movement of the weight. This provides a ready means of changing the speed of the engine without any change in the initial tension or strength of the leaf spring, or change of the governor weight; shifting the point of attachment of the link toward the eccentric arm fulcrum increases the closeness of the regulation.

The speed variation obtainable by shifting one end of the governor spring is usually quite small; any large variation in engine speed requires change of the governor weights and also a new spring or new springs.

53. The adjustment of a shaft governor for speed and sensitiveness is a matter of trial, noting the behavior of the engine before and after an adjustment has been made. Before attempting to adjust a shaft governor which does not function satisfactorily, it is well to inspect the valve and valve gear for freedom of motion, as a steam valve that moves very hard, as for instance on account of the valve-stem packing having been tightened too much, or a binding in any part of the valve gear, will greatly interfere with the governor action. Furthermore, the governor itself should be inspected to see that all parts can move freely; if the pivot or pivots of the governor or weights have run dry and cut so as to give very large friction, the governor action will be very irregular, the speed changes not being definite at all times under the same change of load. All defects of the valve, valve gear, and governor action should be corrected before changing any adjustment of the governor

MISCELLANEOUS GOVERNORS

PRESSURE GOVERNORS

54. A **Ford** regulating valve, which is a device that tends to maintain a fairly constant pressure in a tank by stopping or starting a steam pump delivering into the tank, is shown in Fig. 29. It consists of a spring-actuated steam valve *U* and a water piston *V* moving in a little cylinder *W* under the influence of the water pressure, the water being admitted at *p*. By properly adjusting the spring, the steam valve can be made to close when the water pressure on the piston *V* exceeds a certain required amount. The regulating valve is placed in the steam supply pipe to the pump in a vertical position between the steam chest and an ordinary throttle valve. The lubricator for the pump is placed so as to allow the oil to pass through the regulating valve. The pipe connecting the pressure tank with the regulating valve is provided with a globe valve and a union next to the valve, in order that the cap may easily be removed for repacking the piston *V*. A drip pipe is connected with the bottom of the cylinder *W*.

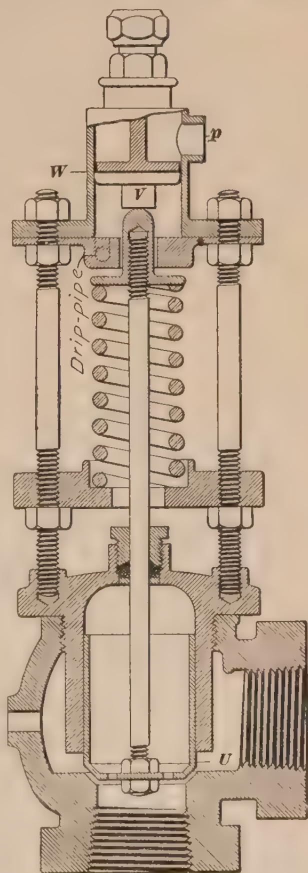


FIG. 29

55. A **Mason** pressure regulator or pressure governor as applied to steam pumps is shown in Fig. 30 in section and in perspective in Fig. 31. Referring to the illustrations, the operation is as

follows: Steam from the boiler enters the regulator at the point marked *Inlet* and passes through into the pump, which continues

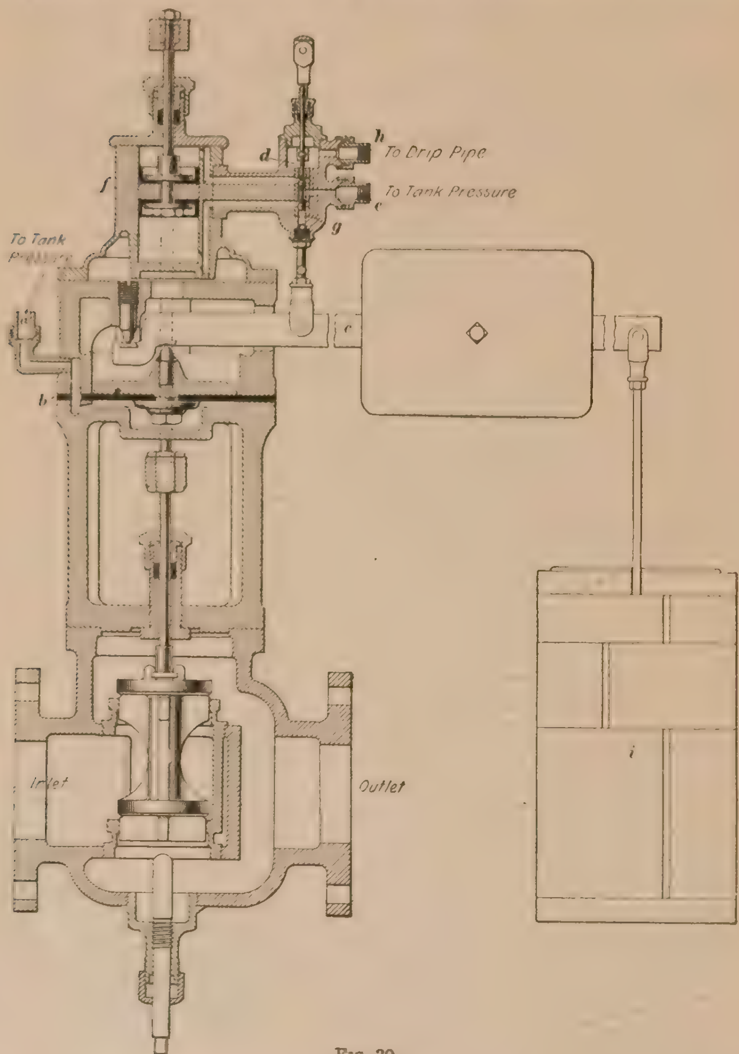


Fig. 30

in motion until the required water pressure is obtained in the tank, which pressure acts on the diaphragm *b* through a $\frac{1}{4}$-inch

pipe connected at *a*. This diaphragm is raised by the excess water pressure and carries with it the weighted lever *c*, opening the auxiliary valve *d* and admitting the water pressure from the connection *e* to the top of the piston *f*. At the same time the exhaust port under the piston *f* is opened, thus allowing the water under this piston to escape through the passage *g* shown in dotted lines, into the drip pipe *h*, thereby pushing down the

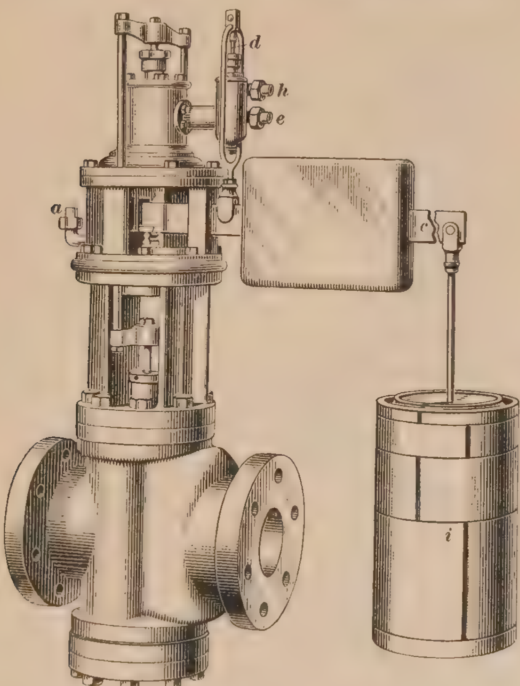


FIG. 31

piston, which closes the steam valve and stops the pump. As soon as the pressure in the system is slightly reduced, the lever *c*, on account of the reduced pressure under the diaphragm, is forced down by the weights *i*, carrying with it the auxiliary valve *d* and thus opening the exhaust from the top of the piston, and at the same time admitting the water pressure under this piston, which is now forced up and opens the steam valve, again starting the pump.

WATER-LEVEL GOVERNORS

56. To obtain the best service from steam boilers, the supply of boiler feedwater should be continuous, automatically regulated according to the demands on the boilers, and the

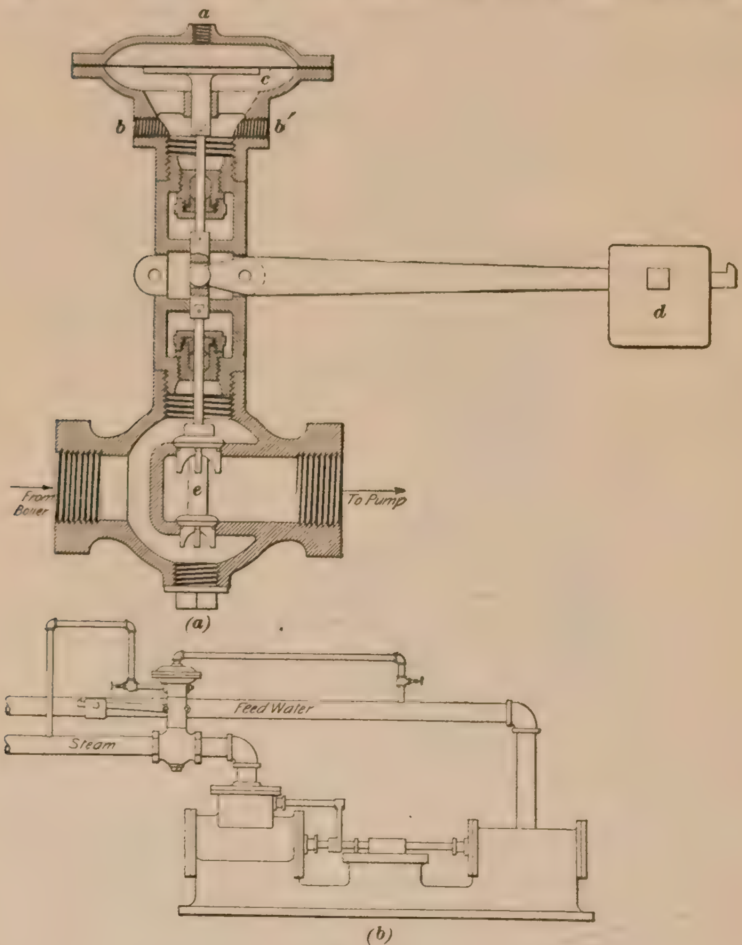


FIG. 32

water-line in the boilers should be maintained at a virtually constant level. This is usually accomplished by means of a

pump governor actuated by changes in water level, which produce differences of hydraulic pressure. In order to give satisfactory service, a pump governor that is intended to maintain a fairly constant water level in steam boilers, must not be affected by variations in steam pressure.

The **Fischer** constant-excess-pressure governor for boiler feed-pumps, which is shown in Fig. 32, as implied by the name,

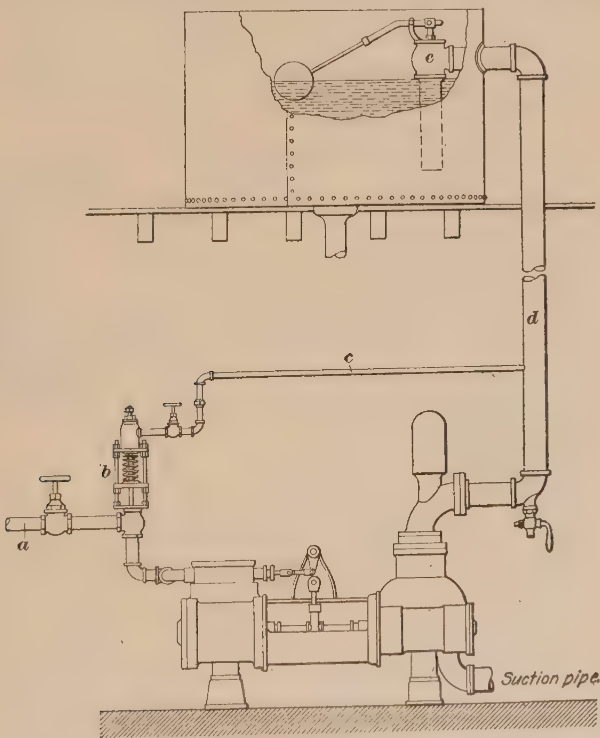


Fig. 33

maintains a pressure in the boiler feed line that is always in excess, by the same amount, of the boiler pressure regardless of its variations. Connection to the feedwater pipe is made at *a*, view (*a*), and to the steam pipe at *b* or *b*′. The steam is admitted, through suitable openings, to the under side of the diaphragm *c*, and the water pressure acts on the upper side.

By means of the weight *d*, the governor can be so adjusted that when a certain level is maintained the throttle valve *c* will be closed and the pump stopped. Both the upper and lower sides of *c* are connected to the boiler, the upper side connecting to the water space (through the feedpipe) and the lower side to the steam space. A change in water level will therefore affect the resultant pressure on the diaphragm. If the level is lowered, the pressure on the upper side becomes less and the steam pressure raises the valve *c*, thus starting the pump, which continues to operate until the downward pressure on *c*

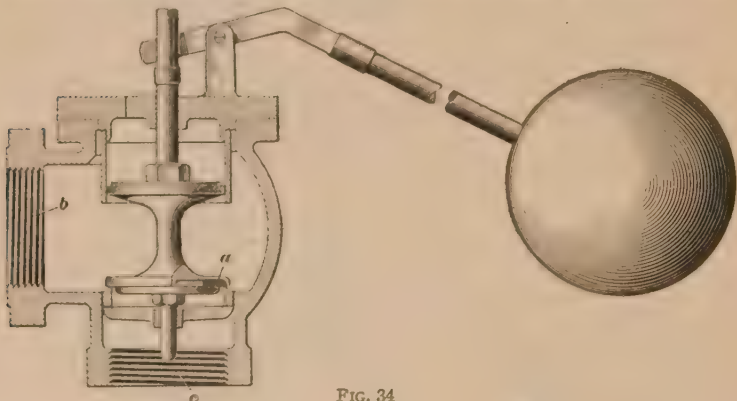


FIG. 34

overcomes the upward steam pressure and closes the valve *c*, thus stopping the pump. Fig. 32 (*b*) shows the method of piping the governor.

57. A common method of governing a steam pump to maintain a virtually constant water level in an open tank is shown in Fig. 33. The steam pipe *a* is fitted with the regulating valve that was shown in Fig. 29, and this is connected by the pipe *c* to the pump discharge pipe *d*. The discharge pipe is fitted, in the tank, with a balanced float valve *e* that closes when the tank is full of water, thus compressing water in the pipe. This pressure in the pump discharge pipe is transmitted through the pipe *c* to the regulating valve, which it closes, thus cutting off the supply of steam and stopping the pump. When the water level in the tank is lowered, the float falls with it,

thus opening the balanced valve and relieving the pressure on the regulating valve, which is opened immediately by a spring and starts the pump again.

58. A sectional view of a balanced float valve is shown in Fig. 34. The valve, as the name implies, is balanced and closes with the pressure of water. It is provided with a soft disk seat *a* that effects a perfect seal when the valve is closed. The inlet port of the valve is at *b* and the discharge outlet at *c*.

VALVE GEARS.

(PART 1.)

DEFINITION AND CLASSIFICATION.

DEFINITION.

1. The **valve gear** of an engine may be defined as that part of the mechanism which acts to control the distribution of steam to the cylinder. In the plain slide-valve engine, the valve gear consists of the eccentric, eccentric rod, rocker-arm or valve-stem guide, valve stem, and the valve. In the different types of reversible engines, or variable and automatic cut-off engines, the valve gear includes the mechanism by means of which the motion of the valves is governed so as to change the direction of motion of the engine or vary the amount of work done in the cylinder.

GENERAL CLASSIFICATION.

2. Valve gears are classified in a variety of ways, depending partly on their type and partly on the kind of service for which they are used. The most common terms used to denote the different types and classes of service are the following:

3. Fixed Cut-Off Gears.—In this type the motion of the valve does not vary with the amount of work to be done in the cylinder. With a given setting of the valve, cut-off

is constant. A plain slide-valve engine is the most familiar example of an engine fitted with a fixed cut-off gear. With it the points of admission, cut-off, release, and compression do not vary.

4. Variable Cut-Off Gears.—Under this head are included all those gears in which the amount of work done in the cylinder can be regulated while the engine is in operation by varying the motion of the valve so as to change the point of cut-off. If the motion of the valve is controlled automatically by some form of governor, a variable cut-off gear becomes an **automatic cut-off gear.**

5. Positive-motion gears are those in which the valve is directly controlled by the eccentric and governing mechanism throughout its whole range of motion.

In **cam gears** the valves receive their motion from cams instead of eccentrics. Cam gears are much used in connection with poppet valves.

6. Releasing gears, or **trip gears,** as they are sometimes called, vary the point of cut-off by releasing the admission valve from the control of the eccentric, and thus permit it to be quickly closed by some other device, usually a spring or dashpot. The Corliss valve gear is the most familiar type of releasing gear.

7. Reversing gears are used to change the direction of motion of the engine; in most cases they may also be used to vary the point of cut-off; hence, most reversing gears are also variable cut-off gears. There are two principal types of reversing gears, viz., link motions and radial gears.

8. The classification may be further extended to make it cover the number and type of valves used to control the steam distribution. In accordance with the number of valves, the leading divisions are **single valve, double valve,** and **four valve.** The leading types of valves are **slide valves, rotary valves,** and **poppet valves.**

CUT-OFF, OR EXPANSION, VALVES.

9. Purpose of Cut-Off Valves.—In order to extend the range of cut-off beyond that which may be obtained with the plain slide valve driven by a fixed eccentric and at the same time not affect the events of admission, release, and compression, various types of auxiliary valves, generally known as **expansion valves,** or **cut-off valves,** have been designed for use in conjunction with the plain slide valve. These auxiliary valves are generally driven by a separate eccentric; their sole purpose is to stop the flow of steam to the cylinder at the desired point in the stroke, leaving the other features of steam distribution to be controlled by the main valve in the usual manner. Since the point of cut-off is the only event controlled by the auxiliary valve, any change in the proportions of this valve or in the position of its eccentric that will secure the desired change in cut-off may be made without affecting the other events in steam distribution; consequently, the point at which the auxiliary valve cuts off the supply of steam may be varied in any convenient way; for example, the lap of the valve or the angle of advance or eccentricity of the auxiliary eccentric may be changed as may be most convenient.

10. A simple application of the cut-off valve is shown in Fig. 1. The main valve V is a plain slide valve driven by a fixed eccentric in exactly the same manner as in the plain slide-valve engine. This valve is given such proportions as will secure the desired points of admission, release, and compression, care being taken that the lap is such that it will not cut off steam earlier in the stroke than the latest point at which it may be desired to have cut-off occur, usually about $\frac{3}{4}$ stroke.

The main valve is located in a section S of the steam chest that is separated from the remainder by a plate or partition p. This partition has ports or openings t, t that form passages connecting the two chambers, and it also forms the valve seat of the auxiliary valve v. The auxiliary

valve has ports *s*, *s*. It is driven from the main shaft by an auxiliary eccentric. When it is in mid-position, as shown in the figure, the ports *t*, *t* are open and allow steam to

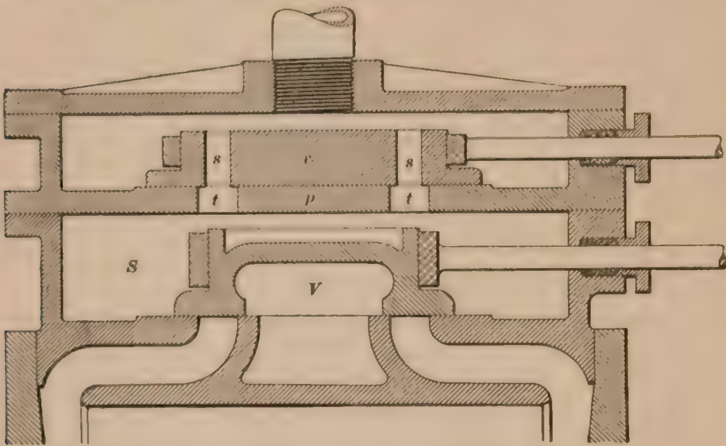


FIG. 1.

fill the chamber around the main valve. As it moves either way from the mid-position, it covers the ports *t*, *t*, and so shuts off the supply of steam from the main valve.

11. The main valve is set in exactly the same manner as any plain slide valve. The auxiliary valve must be so proportioned and set that it will open the ports *t*, *t* as soon as or before the main valve opens the main port, but they must not be reopened before the cut-off occurs with the main valve, otherwise steam will be readmitted to the cylinder and the effect of cut-off will be destroyed. The point of cut-off may be varied by changing either the eccentricity or angle of advance of the auxiliary eccentric, the latter being the more common method. If, as has sometimes been done, but one port *t* is used, and the valve is made of two plates, the cut-off may be varied by changing the distance between the edges of the plates forming the auxiliary valve. By either of the first two methods, the practical range of cut-off is small; by the third, a greater range can be obtained. A disadvantage of this type of expansion valve is that when

cut-off takes place, the space in the main-valve chamber is filled with steam. This has the effect of increasing the clearance space during the period of expansion, and thus destroys some of the benefits derived from the earlier cut-off. On account of these limitations and disadvantages, this simple form of expansion valve which, from the name of its designer, is known as the **Gonzenbach valve,** is but little used to-day.

12. The **Meyer cut-off valve,** Fig. 2, is an improvement on the preceding type and is free from most of its dis-

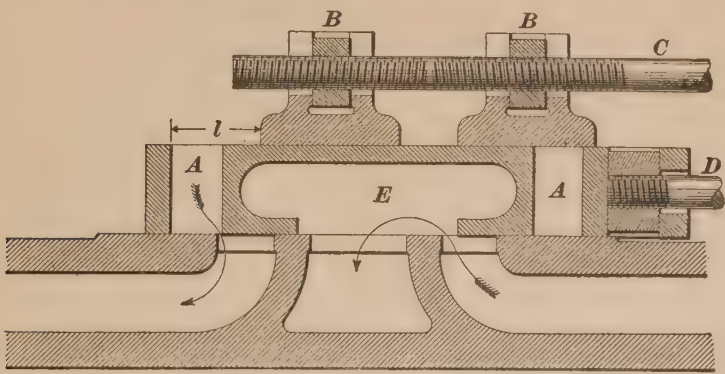


FIG. 2.

advantages. The main valve consists of a flat plate with a chamber *E* on the under side. It acts in exactly the same manner as a plain slide valve, from which it only differs in that the ends of the valve are extended so as to form passages or ports *A*, *A* leading to the edges that control the admission of steam to the cylinder. The cut-off valve consists of two plates *B*, *B* that slide on the top of the main valve so as to close the passages *A*, *A* at the point at which it is desired to have cut-off take place. The main valve is driven from its eccentric by the valve stem *D* and the cut-off valve from a separate eccentric by the valve stem *C*. The main eccentric is set in the same manner as with the plain slide valve. The auxiliary eccentric is so set that, when cut-off takes place, the cut-off valve has a motion opposite in direction to that

of the main valve. The cut-off valve thus closes the passage *A* through which steam is being admitted to the cylinder. The action may be studied by considering Fig. 3 in connection with Fig. 2.

13. In Fig. 3, *O a* represents the crank in the dead-center position, corresponding to the valve position shown in Fig. 2, while *O b* and *O c* represent the corresponding positions of the main and auxiliary eccentrics. It will be noticed that the main eccentric *O b* is set in its usual position of 90° + angle of advance ahead of the crank, thus giving the main valve the amount of lead shown in Fig. 2. The auxiliary eccentric *O c* is set in a position nearly opposite to that of the crank. By inspecting the two figures, we see that as the crank moves in the direction shown by the arrows, the main valve, driven by the eccentric *O b*, will move to the right and open the left-hand port; at the same time the cut-off valve, driven by the eccentric *O c*, will move to the left and it will close the passage *A* as soon as the combined motion of the two valves is equal to the distance *l*, Fig. 2.

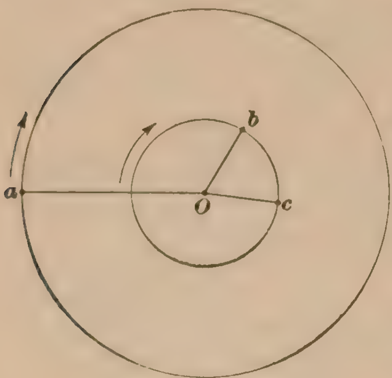


FIG. 3.

The valve stem *C* is provided with right- and left-hand threads that work in threaded plates or nuts in the valve plates *B*, *B*. By turning *C* in one direction or the other, the distance between the outside edges of the plates *B*, *B* can be increased or diminished. If the distance between the plates is increased, the distance *l* becomes shorter and cut-off takes place earlier; the opposite motion makes cut-off take place later. As usually arranged, the rod *C* may be turned by a hand wheel placed outside of the steam chest, so that cut-off may be changed while the engine is running.

14. A form of riding cut-off valve that is somewhat simpler than the Meyer valve is sometimes used. In this form the cut-off valve consists of a single plate, the outside edges of which close the passages through the main valve in the same manner as the edges of the two plates of the Meyer valve. Since the distance between the outer edges of this cut-off valve is constant, the point of cut-off can be changed only by a change in the eccentric, the usual method being to vary the angle of advance.

15. Setting the Riding Cut-Off Valve.—The first step in setting a riding cut-off valve is to examine the main valve to see that it is properly set; that is, try the lead at both ends and set the valve if it is unequal. The cut-off valve should then be so adjusted that it will travel equal distances on each side of its central position, when referred to its seat on the back of the main valve. If the cut-off valve is of the non-adjustable type described in Art. **14,** set its eccentric in the following manner: Turn the engine to the point at which it is desired to have cut-off take place; then turn the cut-off eccentric until the edge of the cut-off valve just closes the passage of the main valve that is open to the cylinder; the eccentric may then be fastened to the shaft.

16. To set the Meyer valve, the first step is to adjust the length of the eccentric rods in the same manner as for the non-adjustable type. The screw of the valve stem should then be turned so as to separate the plates to the greatest distance at which they are designed to be worked; stops are often provided to limit this distance between the plates. To set the eccentric, turn the engine to the point at which it is desired to have earliest cut-off take place, turn the eccentric on the shaft until the cut-off valve just closes the passages through the main valve, then fasten the eccentric to the shaft.

17. The Meyer valve is sometimes used to secure a variable cut-off with a reversing engine; when so used, if the main eccentrics are given equal angles of advance and the cut-off eccentric is set exactly opposite to the crank, the

cut-off will be the same for either direction of motion of the engine. It is, however, sometimes important to secure a certain range of cut-off for one direction of motion that cannot be secured by such a setting of the eccentric. For such a case, the valves are set with special regard to the more important direction of rotation.

A modification of the Meyer valve is shown in Fig. 4. Here the cut-off valve consists of a single plate *a* with inclined

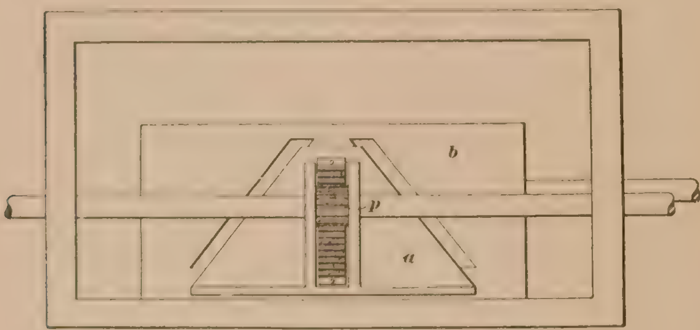


FIG. 4.

edges working over a main valve *b* having passages similarly inclined. The cut-off valve stem, in addition to the endwise motion given to it by its eccentric, may be rotated. It carries a pinion *p* that works in a rack on the cut-off valve. By this means the cut-off valve may be moved across its seat on the main valve; this will have the same effect on the effective distance between the edges of the cut-off valve and the edges of the ports through the main valve as the changes made by the right- and left-hand screws in the distance between the plates of the Meyer valve.

Another somewhat similar modification is made by giving the cut-off valve and its seat a circular form; in this case the ends of the cut-off valve and the openings through the main valve are helical, their shape being the same as though the valve and seat of Fig. 4 were wrapped around a cylinder. By rotating the cut-off valve around its center by means of

the valve stem, the same effect is produced as by sliding the cut-off valve in Fig. 4 across its seat. The turning motion of the valve stem in the two types of cut-off valves here described is often accomplished by a governor, thus producing an engine with an automatic cut-off.

VALVES FOR SINGLE-VALVE AUTOMATIC CUT-OFF ENGINES.

INTRODUCTION.

18. It is a well-known fact that the action of a shaft governor is materially affected by the force required to drive the valve; a plain **D** slide valve, requiring as it does a great driving force, not only absorbs considerable of the power of the engine, but it has a disturbing effect on the action of the governor that, especially with large engines, becomes so serious as to make the use of this type of valve with shaft governors impracticable. To relieve the governor as much as possible from the effects of the frictional resistances incident to the **D** slide valve, a great number of balanced valves, of which the piston valve is one type, have been invented. When used in conjunction with a shaft governor that varies the throw of the eccentric, the plain slide valve has another defect that is not overcome by the mere process of balancing. This defect is that the reduced travel of the valve consequent on the short throw of the eccentric at early cut-offs has the effect of greatly restricting the port opening. At high speeds such a reduction in port opening causes an imperfect filling of the cylinder with live steam; that is, the steam is wiredrawn in its passage through the partly opened ports. This difficulty can be largely overcome by increasing the travel of the valve so as to give it a considerable amount of **overtravel** for late cut-offs. By overtravel is meant that the valve is given a travel greater than is required to give a full port opening of the steam ports. Such a remedy, however, involves other undesirable conditions.

To provide for the overtravel, the central chamber of the valve and the bridges between the steam and exhaust ports must be made wider; this means a larger and heavier valve. The attempt to secure a liberal port opening with early cut-offs by giving the valve overtravel thus involves an increase in the travel of the valve and in its weight, both of which add to the power required to move it. Once during each stroke the heavy valve must also be started from rest and given a velocity that becomes greater as the distance through which the valve must move is increased; to overcome the inertia of the valve, a very considerable force is required, and this force, which must act through the medium of the governor, and in consequence tends to disturb the governor action, is increased by the attempt to secure a liberal port opening by means of an increase in the valve travel.

Several designs for shaft-governor engines secure an increased port opening by the use of multiple-ported valves, in which steam enters the cylinder through two or more passages. While the travel of the valve may be reduced by this means, its size and weight are generally increased, and this neutralizes the advantage of short travel to a considerable degree.

THE PISTON VALVE.

19. The **piston valve** is one of the lightest and most perfectly balanced types that has yet been devised, and it is readily made double-ported. It has, however, several features that have restricted its use to a very great extent. Unless it is provided with some form of packing there is no means of adjustment to provide for wear; consequently, it soon becomes leaky and wastes steam badly. In vertical engines, where the weight of the valve is not supported by the seat, the wear is not so serious, and there are many cases of such engines in which piston valves with no packing except grooves in which water collects have given the best of service. In a horizontal engine, however, the weight of the valve invariably wears the lower part of the seat and

destroys its circular form. Packing rings of some form must here be provided or the valve will soon leak. In some cases the valve seat consists of a bushing that can easily be removed when worn; in order that a good fit between valve and bushing may be secured, it will generally be necessary to renew the valve as well as the bushing, or to fit the old valve and new bushing to each other. The long ports required to surround the piston valve make the clearance large. This type of valve as generally constructed offers no relief to water that may be caught in the compression space after the exhaust port closes.

THE FLAT PRESSURE PLATE VALVE.

20. The **flat-pressure plate valve** is a type that has many of the advantages of the piston valve and at the same time overcomes some of its faults. Fig. 5 shows a form of this valve used in the engines built by the Ames Iron Works, and is a form that, with several minor modifications, is used in a great number of shaft-governor engines. The valve V consists of a thin rectangular casting with openings or ports through it for the passage of live and exhaust steam to and from the cylinder. It works between the face of the valve seat and a similar face formed by a heavy pressure plate P. The pressure plate is separated from the valve seat by strips s, s' that are made just enough thicker than the valve to permit the valve to slide freely between the pressure plate and the valve seat. When the engine is working, the pressure plate is held in place against the strips s, s' by the steam pressure; when steam is shut off, the spring N prevents the plate from falling away from its position against the strips. This construction permits the valve and pressure plate to act as a relief valve for the escape of water from the cylinder; an excessive pressure in the cylinder will lift the valve and plate and compress the spring. The danger of breakage from water in the cylinder is thus somewhat reduced; it is not, however, a perfect protection; with quick rotative speeds, the pressure often

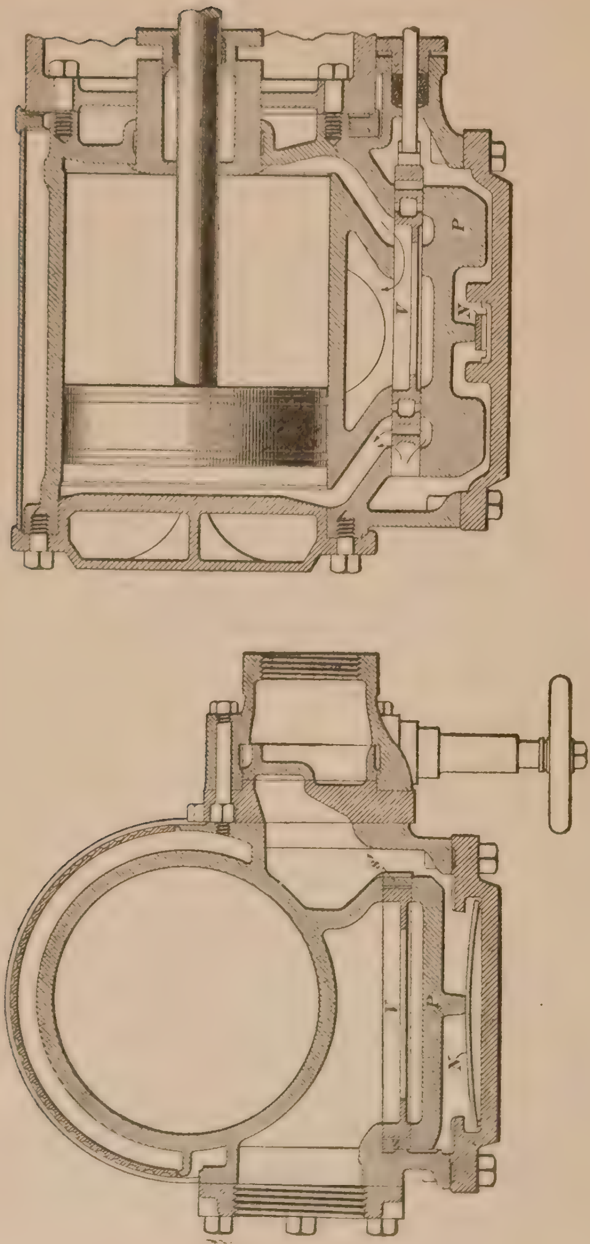


FIG. 5.

becomes great enough to seriously damage the engine before water can be forced out through the ports and overcome the inertia of the valve and pressure plate.

21. An advantage of the pressure-plate valve over the piston valve is that in horizontal engines the wear due to the weight of the valve, coming as it does on the lower edge and the strip s', does not affect the tightness of the valve. Another advantage is that when the valve wears so as to allow steam to leak between it and the faces of the valve seat and pressure plate, the strips s, s' can be planed or scraped so as to overcome this wear; this operation, however, requires considerable care and skill. If the valve or the faces between which it works are unevenly worn, it will be necessary to scrape them to a new surface; this is a tedious and somewhat difficult task with the facilities available in most engine rooms.

With either piston valves or pressure-plate valves, the precaution is required when starting the engine to admit steam to the steam chest slowly so as to warm the valve and its seat gradually and evenly; if steam is turned on too rapidly, there is danger that the valve will be heated faster than its seat and swell so much as to bind or seize in its seat and break some part of the valve gearing. To reduce this danger, one make of engine using a pressure-plate valve, the **Woodbury,** is provided with a device for lifting the pressure plate a short distance from the valve until the valve and its seat become uniformly heated.

SELF-ADJUSTING BALANCED VALVES.

22. Methods of Balancing.—Many of the faults of the valves described in Arts. **19** to **21** are overcome by the use of self-adjusting balancing devices, of which there are several types. In one the pressure plate is held against the back of the valve and the valve against its seat by a certain amount of unbalanced steam pressure. In another form that has been much used on locomotives and some types of stationary engines, although not to so great an extent on

shaft-governor engines, the back of the valve is fitted with packing rings or strips that bear against a fixed pressure plate and prevent live steam from reaching the space between the pressure plate and the back of the valve. Fig. 6 shows an

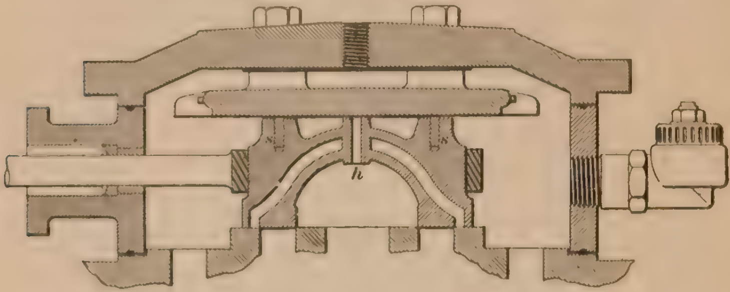


FIG. 6.

Allen valve with this method of balancing. The packing strips *s*, *s* are fitted in grooves in the back of the valve and held out against the pressure plate by light, flat springs. A hole *h* connects the space at the back of the valve with the exhaust cavity; the pressure in this space is, therefore, the same as the pressure in the exhaust port.

23. The **Ball valve,** Fig. 7, is a form of self-adjusting balanced valve that is made of two separate pieces, one working telescopically in the other, the joint between the two being kept steam-tight by packing rings. Each part of the valve bears against its own seat and controls the flow of steam through its own set of ports. Live steam enters the chamber *a*, formed by the combination of the two parts, through the opening in the upper wall of the steam chest and passes through the ports to the cylinder; the exhaust is discharged into the steam chest around the outside edges. The valve is thus seen to be *indirect* and the combination of the two parts, each with its separate ports, secures a double-ported effect for both steam and exhaust. The two parts of the valve are held against their respective seats by the pressure of the steam in the chamber *a*, the action of this pressure tending to separate the parts. By this arrangement

the valve automatically follows up any wear, and if the working faces are kept in good order it will always be tight. It is not quite as perfectly balanced as the piston valve or the flat-pressure plate valve of the type described

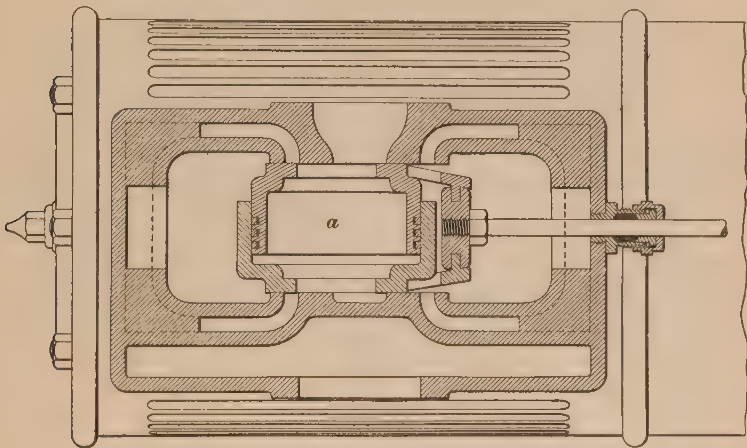


FIG. 7.

in Arts. **20** and **21**, and its weight is probably a little greater than either of these types. With it the ports are necessarily long and the clearance space is large; in this respect it is probably to be compared with the piston valve.

THE CORLISS VALVE GEAR.

CHARACTERISTIC FEATURES.

24. The most characteristic features of the Corliss valve gear are four in number :

1. *The Four Oscillating Valves.*—Two of these valves are placed at each end of the cylinder—one (the admission valve) controlling the admission of the steam, the other (the exhaust valve) controlling the liberation of the expanded steam.

2. *The Wristplate.*—This is an oscillating plate, disk, or lever actuated by the eccentric on the main shaft and in turn communicating motion to the four valves.

3. The *releasing mechanism*, of which the governor forms a part. This disconnects an admission valve from the wristplate when the supply of steam is to be discontinued through its port.

4. *The Closing Apparatus.*—This, immediately upon the release of an admission valve, brings the valve back to its closing position.

Corliss valve gears have been built in a great variety of forms, differing from one another principally in the details of the releasing mechanism and in minor features of construction. These gears, however, all operate on the same general principles, and when one is understood there will be little difficulty in understanding any of the others.

CONSTRUCTION.

25. Cylinder and Valves.—Fig. 8 is a section that shows the general construction of one form of cylinder and valves. As shown, there are four distinct valves. Two of these, v' and v, admit steam to the crank end and head end of the cylinder, respectively. They are in communication with the steam chest d and steam pipe s. The other two valves, r and r', communicate with the exhaust chamber l and exhaust pipe o. The valve r' allows the steam to exhaust from the crank end of the cylinder; r controls the exhaust from the head end.

The valves v, v' are called the **steam valves,** and r, r' are called the **exhaust valves.**

The valve faces are cylindrical in form, and the valves rotate on circular valve seats. Each valve extends across the cylinder; that is, the length of the valve is about equal to the cylinder diameter.

An inspection of Fig. 8 shows that the steam valves open their ports when their faces move away from the center of

the cylinder towards the heads. It is also seen that both the steam and the exhaust valves are *single-ported;* that is, they

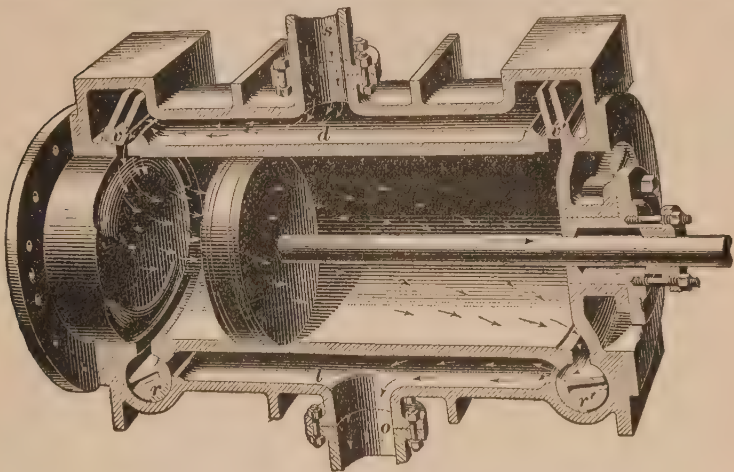


FIG. 8.

open but one passage for the flow of the steam to and from the cylinder.

26. Double-Ported Valves.—Fig. 11 shows a method of construction by means of which the valves are all made double-ported. The steam valves in this case open their ports by rotating in the opposite direction to that in which the steam valves of Fig. 8 rotate; that is, to open the ports the valve faces move away from the heads towards the center of the cylinder. The form of the valves is such that steam flows past two edges, as shown by the small arrows around the steam valve *a* and the exhaust valve *e'*. This construction provides a sufficient port opening with a much smaller valve movement and greatly improves the distribution of steam to the cylinder.

MECHANISM FOR OPERATING THE VALVES.

27. Fig. 9 shows the general arrangement of one of the most familiar forms of the mechanism for operating the valves. The steam valves are driven by the spindles *V*

and *V'*, called the **steam-valve stems**, and the exhaust valves by the spindles *R* and *R'*, called the **exhaust-valve stems**; the center lines of these spindles coincide with the center lines of their respective valves.

To the steam-valve stems *V*, *V'* are keyed cranks, as *N*. In the figure, the crank keyed to *V'* is removed to show more clearly the disengaging link and hook *I'*. Likewise, to the exhaust-valve stems *R*, *R'* are rigidly keyed the cranks *M*, *M'*.

The disk, or **wristplate**, *A* is made to rock on the stud *C* by the wristplate rod *B*, which takes its motion from a rocker-arm that in turn is driven through an eccentric rod by an eccentric placed on the main shaft. To the plate *A* are connected the four **valve rods** *E*, *E'*, *F*, and *F'*. As the plate rocks, these valve rods are given a to-and-fro motion.

28. The exhaust-valve rods *F* and *F'* are connected directly to the cranks *M* and *M'* of the exhaust valves. The motion of the plate *A* is thus communicated to the exhaust valves, causing them to open and shut the exhaust ports at the proper time. The steam-valve rods *E*, *E'*, however, are not connected directly with the cranks keyed to *V* and *V'*, but with the bell-cranks *H*, *H'*, called the **admission cranks.** These bell-cranks can rotate around the valve stems *V*, *V'*, not being connected rigidly to them. The bell-cranks carry the disengaging links *I*, *I'*, which, therefore, have the rocking motion given to the bell-cranks by the rods *E*, *E'*. These disengaging links hook on to blocks *B*, *B'*, rigidly fastened to the cranks keyed to *V* and *V'*. Consequently, when the bell-cranks *H*, *H'*, and with them the links *I*, *I'*, are rotated by the motion of the valve rods, the cranks, as *N*, are forced to move with them. This motion of the crank rotates the valve and admits steam to the cylinder.

29. Cut-Off Mechanism.—On the valve stems *V* and *V'* are collars *G* and *G'*, called the **trip collars**; these collars are free to rotate on the stems. Each collar is provided with

an arm to which is attached a rod UX, called a **governor rod,** that leads to the governor. Each collar also has a projection, as a', called a **trip,** that lies in the same plane as the arms I and I' of the disengaging hooks. Considering now the right-hand admission valve, imagine the wristplate A to be rotating towards the left, in a direction opposite to the direction of motion of the hands of a watch; this motion will be transmitted by the rod E' to the crank H' and will carry the link and hook I' in an upward direction, thus lifting the block B' and rotating the right-handed crank to which B' is attached, which, it will be remembered, was removed to show the other parts more clearly. This crank, being keyed to the stem V', turns this stem and so opens the valve, the direction of motion of the valve face being the same as with the valves described in Art. **26.** As the link I' is lifted, its arm strikes the trip a' on the collar G'; this swings the link I' about the pin by means of which it is pivoted to the crank H' and thus detaches the hook from the block B' and stops the opening motion of the valve. To the cranks keyed to V and V' are attached rods leading to the pistons of the dashpots P and P'. When the crank is rotated by the upward motion of the block B', as just described, the dashpot rod attached to it lifts the piston in P' and creates a partial vacuum in the space below the piston. When the block B' is released from the hook on I', the pressure of the atmosphere forces the dashpot piston downwards, thus quickly closing the valve and cutting off the supply of steam to that end of the cylinder.

When the wristplate A moves in the opposite direction to that just considered, the hook I' is moved downwards until it again engages the block B' and the motion above described is ready to be repeated upon the reversal of the wristplate. The action of the valve for the crank end of the cylinder is precisely the same as for the head end, except that it is opened when the wristplate rotates towards the right.

30. Analysis of the Wristplate Motion.—As will be seen by a study of Fig. 10, the wristplate modifies the effect

of the eccentric motion in such a way that the motion of the valves is considerably improved. In the figure, O represents

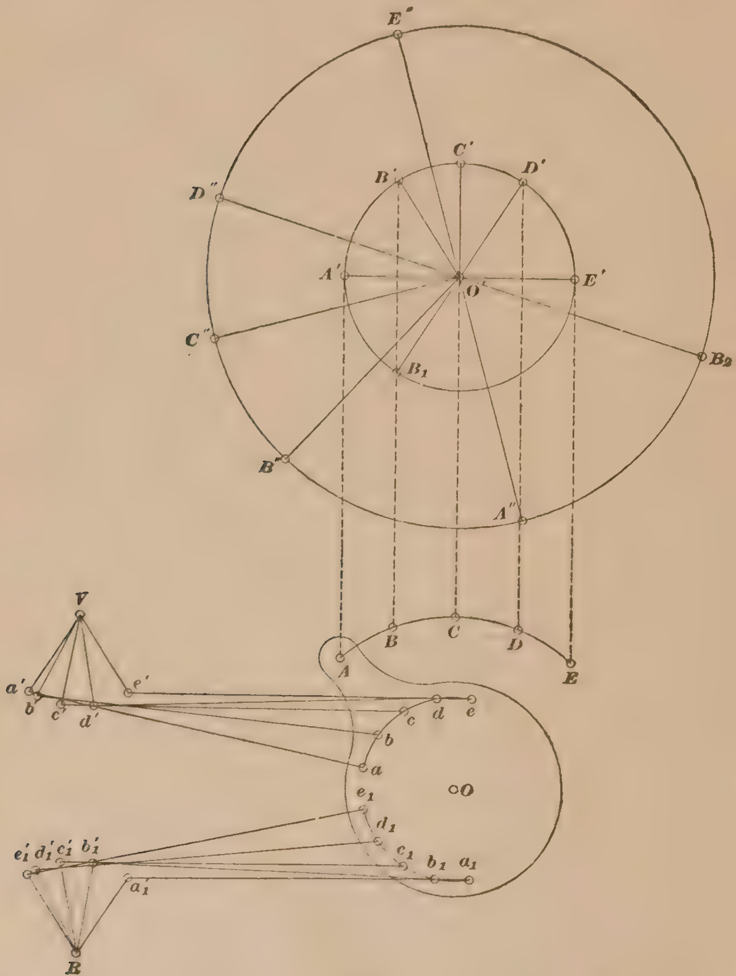


FIG. 10.

the center of the wristplate stud and V and R represent, respectively, the centers of the steam- and exhaust-valve

stems for one end of a Corliss engine gear similar to that shown in Fig. 9. A, B, C, D, and E represent five positions of the pin on the wristplate to which the carrier rod is attached; of these positions, A and E are the extreme positions of the pin and the others are so chosen that the angular distances between two consecutive spaces are all equal. The positions of the steam-valve rods corresponding to the positions A, B, C, D, and E are represented, respectively, by the lines $a\,a'$, $b\,b'$, $c\,c'$, $d\,d'$, and $e\,e'$, and the similar positions of the exhaust-valve rods by $a_1\,a_1'$, $b_1\,b_1'$, $c_1\,c_1'$, $d_1\,d_1'$, and $e_1\,e_1'$. It is clearly shown by the figure that the motion imparted to the valves for a given angular motion of the wristplate varies through wide limits. For example, when the wristplate pin moves from A to B, the steam valve moves through the very short distance represented by $a'\,b'$; when, however, the wristplate pin reaches the middle of its motion, the motion imparted to the valve is much more rapid, as is seen by comparing the distances $C\,D$ and $c'\,d'$ with $A\,B$ and $a'\,b'$.

31. In order to compare the valve motion with the motion of the eccentric and crank, the eccentric and crank positions corresponding to the wristplate positions from A to E are drawn with a center O' vertically over the center O of the wristplate stud. The eccentric center is represented in the six positions, B_1 to E', of which A' and E' are the extreme positions of its throw and the corresponding crank positions are represented by the points on the crankpin circle lettered from B_2 to E''. Beginning now with the crank position B_2 and following the crank around in the direction of the arrow, it is seen that while the crank moves from B_2 to B'', a motion that corresponds to a motion of the piston through nearly $\frac{3}{4}$ of its stroke, the end of the valve crank has moved only from b' to its extreme position a' and back to b' again; during this large portion of the piston stroke the valve has been almost at rest. As the crank passes the position B'' and passes its dead center, the motion of the valve is seen to increase rapidly. This has the effect of opening the valve rapidly just as the piston reaches the

end of its stroke, thus permitting a free admission of steam. The valve continues to open at a good rate of speed until it is closed by the releasing gear or until the eccentric reaches its extreme throw position E'.

32. By comparing the motion of the steam and exhaust valves, it is seen that the period during which the steam valve is nearly at rest corresponds to the period when the motion of the exhaust valve is most rapid. The exhaust valve was opened wide before the piston began its stroke to the left, and the period of most rapid closure is that during which the crank passes B'', corresponding to the point in the stroke at which compression begins. Following the point of exhaust closure, the motion of the valve is very small until the piston approaches the opposite end of its stroke, where release is to take place.

33. Varying the Point of Cut-Off.—An inspection of Fig. 9 shows that the point at which the block B' is released from the hook on I', and, in consequence, the point at which cut-off takes place, depends on the position of the trip a'. If the collar G' is rotated towards the left so as to move a' downwards, the arm of I' in its upward motion will strike a' and thus release the block B' earlier; the opposite motion of the collar will evidently have the contrary effect. We thus see that the point of cut-off can be varied by swinging the collars G and G'. The position of the collars is generally controlled by a pendulum governor acting through the rods $U X$. When the rods are moved in the direction of the arrows X, the trips are lowered and cut-off takes place earlier; motion in the direction of the arrows U raises the trips and causes a later cut-off.

34. Limitations to Range of Cut-Off.—When the engine is working under a light load, the governor collar is kept in such position by the governor that the cut-off trip releases the hook early in its motion, while for a heavier load the reverse is the case. As, however, the motion of the hook is an oscillating one, it can evidently not be released by the trip after it has started its return stroke, and, there-

fore, if the load on the engine becomes so heavy that the governor moves the trip so far that it will not be struck by the hook before the reverse motion of the hook begins, release of the hook will not take place at all. In this case the admission valve will be closed as if positively connected to the admission crank, the same as the exhaust valves.

35. Action of the Dashpot.—As shown in the sectional view, Fig. 9, the dashpot consists of two concentric cylinders of different diameters. In the upper and larger of these cylinders fits a piston having a central projecting portion that extends downwards and forms a plunger that fits the lower and smaller cylinder. The upper cylinder has an opening in its end, not shown in the figure, that may be regulated by a valve or cock. This opening is placed a short distance above the lower edge of the piston when it is in its lowest position; it allows a small quantity of air to enter as the piston rises, and this air checks the downward motion to an extent depending on the amount of opening of the valve. After the lower edge of the piston descends until the opening is covered, the further escape of air is prevented and the air remaining in the cylinder acts as a cushion that prevents the piston from striking the bottom too violently. If the piston descends too rapidly and strikes the bottom of the cylinder, or "slams," it indicates that so much air escapes that not enough is caught under the piston to cushion it properly; this difficulty can be remedied by reducing the opening of the cock. The lower cylinder is made as nearly air-tight as possible; to permit of the escape of any air that may leak in around the plunger, this cylinder should be provided with a check-valve. The escape of air from the opening in the side of the cylinders makes a hissing sound that may be prevented by connecting the opening to a pipe that leads outside of the engine room. Some dashpots are made with an air chamber that forms a reservoir from which air is drawn by the ascending piston and into which the air is discharged again as the piston descends. This prevents the hissing sound without the necessity of using the pipe.

RELATIVE MOTION OF PISTON, CRANK, AND VALVES.

36. Fig. 11 shows the piston nearing the end of its backward stroke and all the valves closed. The wristplate is in

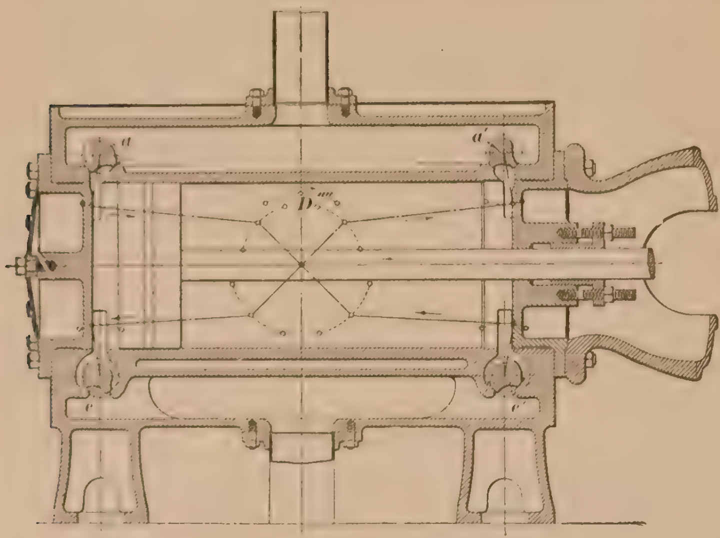
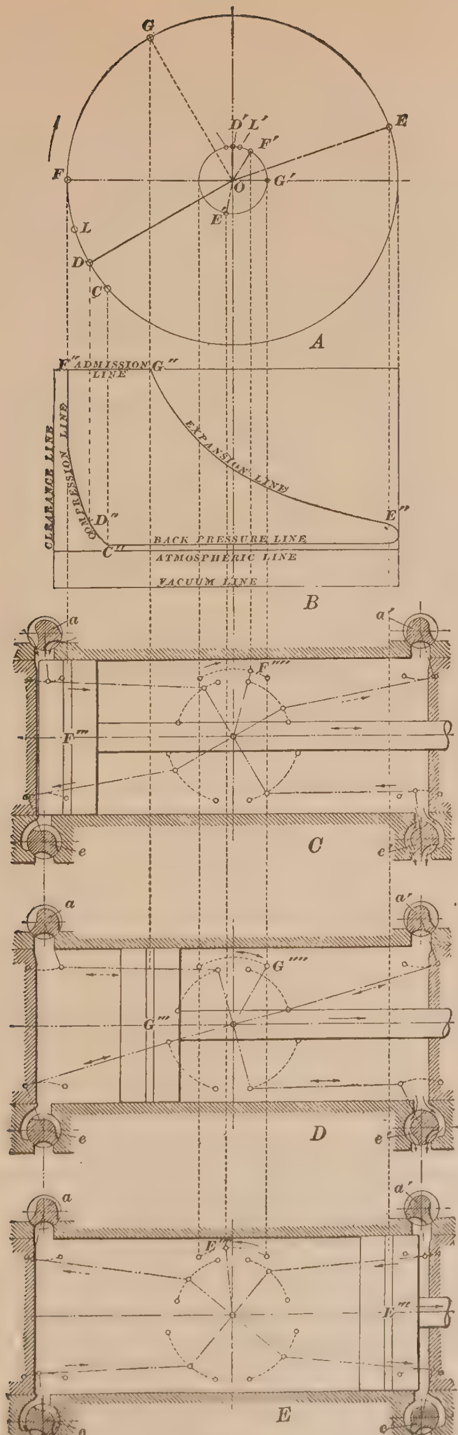


FIG. 11.

its middle position; hence, the exhaust valves are in precisely the same position, relatively, to their respective ports as would also be the two admission valves were it not that the one *a'* is in the released condition. Observing the arrows on the various valve rods, it will be seen that the exhaust valve *e'* will soon be opened to liberate the steam that is still exerting an expansive pressure in the direction of motion of the piston, while the other valve *e* has but lately been closed; the admission valve *a* will also soon be opened to admit steam against the motion of the piston.

37. The diagrams *A*, *B*, *C*, *D*, and *E* of Fig. 12 exhibit the most significant simultaneous positions of the piston and the four valves, together with a skeleton outline of the principal members of the mechanism in their various corresponding positions, during a little more than a complete forward stroke of the engine.



ADMISSION LINE
CLEARANCE LINE
COMPRESSION LINE
EXPANSION LINE
BACK PRESSURE LINE
ATMOSPHERIC LINE
VACUUM LINE
A
B
C
D
E

Fig. 12.

In all the diagrams the directions of motion of the piston, wristplate, and valve rods are indicated by arrows, a double-headed arrow being shown when a member is in the position in which its direction of motion is being reversed, and a dotted arrow being shown when an admission valve rod is moving without affecting its valve, on account of their connecting parts being in the released condition.

At *A*, in Fig. 12, the respective positions of the crank and eccentric that correspond to the positions in Fig. 11 are *D* and *D'*, and the position *C* of the crank indicates when the closing of the exhaust valve *e* actually took place, and the resulting compression commenced, as indicated at *C''* on the diagram *B*, Fig. 12, while the position *L* indicates where the admission valve *a* will open—in other words, the lead position of the crank. Diagram *C*, Fig. 12, shows the piston at the end of its backward stroke; or, what is the same thing, at the beginning of its forward stroke. By this time both the exhaust valve *e'* and the admission valve *a* have been opened considerably, without, however, reaching the limits of their opening positions, while the exhaust valve *e* has nearly reached the limit of its closing position, and this because the four points of attachment of the valve rods to the wristplate are, as will be seen, so located that the angular closing movements of the valves are very small compared with their angular opening movements.

38. At *D*, in Fig. 12, the wristplate is shown in the position in which its motion is just being reversed by the eccentric, the valve rods being, in consequence, also in the positions in which their motions are reversed, as indicated by the double-headed arrows. According to what we have already learned, this constitutes the limiting position for the releasing of the admission valve *a*, which for that reason is supposed to have just occurred, as indicated by its closed position. As indicated at *G''*, on diagram *B*, Fig. 12, this is then the moment at which in this particular case the expansive reduction of the driving pressure on the piston

commences. The exhaust valve e has at the same time reached the actual limit of its closing position, while e' has reached the limit of its opening position. In this position of its valve rod the admission valve a' is picked up, as indicated by one-half of the arrows being represented in full.

G and G', in diagram A, Fig. 12, are, respectively, the positions of the crank and the eccentric that correspond to diagram D, Fig. 12.

39. In diagram E, Fig. 12, the piston is finally represented in that position, near the end of its forward stroke, for which the exhaust valve e just begins to open for the liberation of the expanded steam, e' having been closed some time previously to produce compression, and the admission valve a' also nearing its opening point for the lead position of the piston, which will be reached presently.

Point E'', in diagram B, Fig. 12, shows the release of the expanded steam due to the opening of e, and points E and E', in diagram A, Fig. 12, represent, respectively, the positions of crank and eccentric corresponding to diagram E, Fig. 12.

SETTING CORLISS VALVES.

40. Necessity of Lead and Lap.—In any valve gear the motion of which is derived from the action of an eccentric set at 90° ahead of the crank, the crank will be on its center when the eccentric is at half throw and the eccentric will arrive at its greatest throw, and the opening motion of the valve will cease when the crank is at half stroke. With releasing gears, like the Corliss, the releasing mechanism must operate, if at all, before this point in the eccentric travel is reached; otherwise, the valve will begin to close positively, and at a speed governed by the eccentric on its return stroke. The Corliss valve gear, when properly constructed, closes the valve positively before the end of the stroke, in case the trip does not act or the dashpot fails to close it; this fact does not seem to be generally known. It

follows, then, that when the releasing mechanism does not operate, a steam valve set edge to edge with the port opening is closed at the moment the piston reaches the end of its stroke. The exhaust valve on that end of the cylinder opens at the same moment, and there is a tendency to blow through. On the other end of the cylinder the exhaust valve closes and the steam valve opens, also at precisely the same moment, giving the steam another chance to blow through. It is essential, then, that the steam valve shall have a definite advance in its closing movements, relative to the opening of the exhaust valve, in order that it may have a safe working lap before the exhaust port is opened, and that it may not open until after the exhaust valve has closed. The exhaust valves must also have an advance relative to the piston movement, in order that there may be prompt release, and that they shall close on the end towards which the piston is moving before it arrives at the end of its stroke. This is necessary to produce sufficient compression to check the motion of the reciprocating parts of the engine.

41. Effects of Lead.—By setting the eccentric at an angle of more than 90° ahead of the crank, we get an earlier opening and an earlier closing of all the valves, relative to the piston motion. Unfortunately, however, the same amount of lead is not wanted for all the valves, more lead being required on the exhaust valves so that they will close early and thus give sufficient compression. But this does not help us to get the steam valve closed previous to the opening of the exhaust valve nor the exhaust closed before admission of steam. The effect, then, of lead, as derived from the advance of the eccentric to an angle of more than 90° from the crank, is to hasten both the opening and the closing of all the valves, as regards the motion of the piston; therefore, it hastens cut-off and limits its range.

42. Amount of Lap Required.—The effect of lap on a valve is to hasten its closing and retard its opening. Here,

again, cut-off is hastened and its range limited. From this it will be seen that when a single wristplate is used there are conflicting conditions in the setting of valves between which a compromise must be effected. No definite rule can be given by which the amount of lap for a valve can be determined. It depends somewhat on the design of the valve and its relative proportions; also on the conditions under which the engine is to work. In all cases the lap increases with the size of the cylinder. The table given below furnishes a fairly reliable guide as to the amount of lap to be given to valves on different sizes of engines:

STEAM LAP AND EXHAUST OPENING FOR CORLISS ENGINES.

Diameter of Cylinder. Inches.	Lap for Steam Valve. Inches.	Exhaust-Valve Opening. Inches.	Diameter of Cylinder. Inches.	Lap for Steam Valve. Inches.	Exhaust-Valve Opening. Inches.
12	$\frac{1}{4}$	$\frac{1}{32}$	30	$\frac{1}{2}$	$\frac{1}{8}$
14	$\frac{5}{16}$	$\frac{1}{32}$	32	$\frac{1}{2}$	$\frac{1}{8}$
16	$\frac{5}{16}$	$\frac{1}{32}$	34	$\frac{1}{2}$	$\frac{1}{8}$
18	$\frac{3}{8}$	$\frac{1}{16}$	36	$\frac{1}{2}$	$\frac{1}{8}$
20	$\frac{3}{8}$	$\frac{1}{16}$	38	$\frac{9}{16}$	$\frac{3}{16}$
22	$\frac{3}{8}$	$\frac{1}{16}$	40	$\frac{9}{16}$	$\frac{3}{16}$
24	$\frac{7}{16}$	$\frac{3}{32}$	42	$\frac{9}{16}$	$\frac{3}{16}$
26	$\frac{7}{16}$	$\frac{3}{32}$	44	$\frac{5}{8}$	$\frac{1}{4}$
28	$\frac{7}{16}$	$\frac{3}{32}$	46	$\frac{5}{8}$	$\frac{1}{4}$

43. Centering Rocker-Arm and Wristplate. — A method of centering the wristplate is illustrated in Fig. 13 by the plumb line $x\,y$ let fall from the hand. Usually, however, it is not necessary to resort to this scheme; in most cases there will be found three marks, as at c, b, d, Fig. 14, on the wristplate bracket and another mark a on the hub of the wristplate, by means of which the wristplate may be

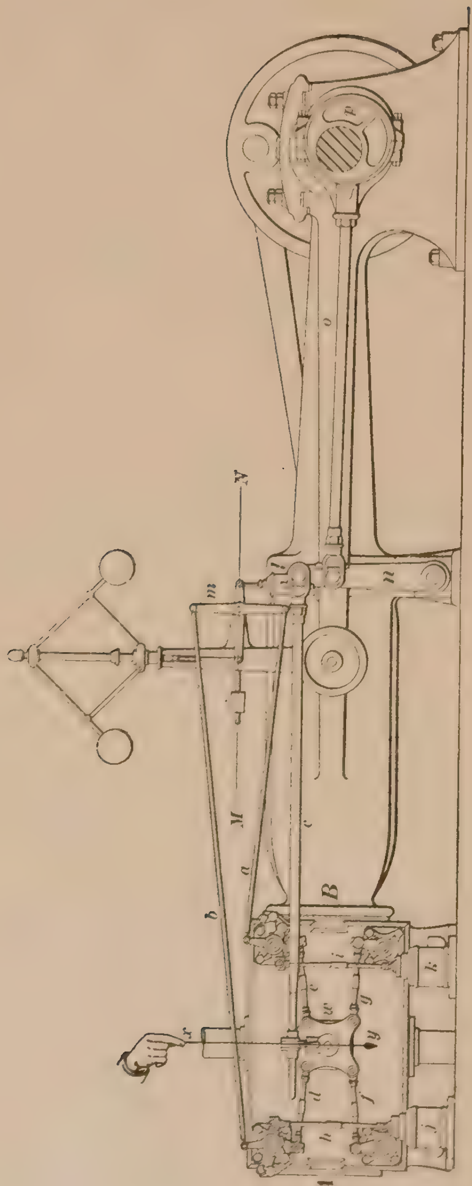


FIG. 13

centered. The marks are so located that *a* is opposite *b* when the wristplate is at its center of motion. At the two extremes of motion *a* is opposite either *c* or *d*.

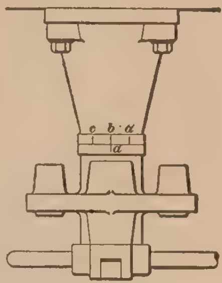


FIG. 14.

It may be well, however, to test these marks, or rather to see that the eccentric and carrier rods have proper adjustment relative to the motion of the wristplate. To do this, rotate the eccentric *p*, Fig. 13, on its shaft, having the eccentric rod *o* connected and the carrier rod *c* hooked over on the wristplate; then notice whether or not the rocker-arm *n* is equidistant in its extreme travel each way from a plumb line let fall through the center of its pin. If it is not, make it so by adjusting the length of the eccentric rod *o*. Next, see if the mark on the wristplate hub agrees with those on the bracket at full throw each way; if not, the remedy is to change the length of the carrier rod *c* until there is perfect agreement. Having tested the marks temporarily, secure the wristplate *w* at its center of motion.

44. Adjusting the Lap.—Upon removing the back bonnets, or caps, from the ends of the valve chambers, so that the rear ends of the valves are exposed, a mark will usually be found on each face of the valve ports, showing the location and width of the port openings in relation to the cylinder. Upon the ends of the valves, marks in line with the opening edges of the valves will also be found. See Fig. 15. Possibly in some of the older types of engines these may be missing; in such a case the valves will have to be removed to locate the port openings and the opening edges of the valves. Consulting the table, Art. **42,** find for the engine being adjusted the lap for the steam valves and the opening to be given the exhaust valves. By lengthening or shortening the rods leading from the wristplate to the valve arms, bring the opening edges of the valves to positions corresponding with the desired lap or

opening. While making these adjustments, it is, of course, essential that the steam latch shall be hooked on the stud.

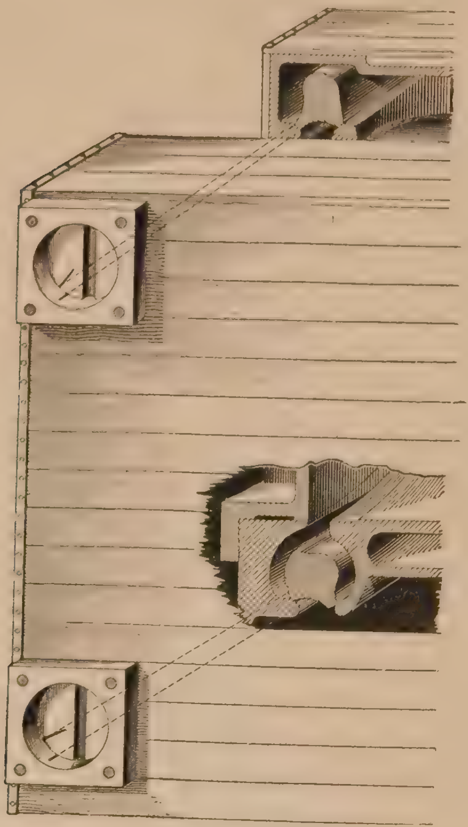


FIG. 15.

If a record is kept of how much the valve moves at one turn of the adjusting nut on the rod, future adjustments may be made without the removal of the bonnets.

45. Adjusting the Lead.—All the valves are now supposed to be in their proper positions when the wristplate is at its center of movements. The section shown in Fig. 11 gives this position, except that a' is supposed to be

unhooked and to stand in a position of full closure. The next step is to locate the eccentric at the proper angle ahead of the crank to give sufficient lead. First, set the engine exactly on its center; and with the carrier rod hooked on the wristplate stud revolve the eccentric on the shaft in the direction in which the engine is to run, until it is at an angle greater than 90° ahead of the crank or until the steam valve on the end at which the piston stands is just beginning to open. In this position the eccentric must be secured to the shaft. Then turn the engine to the other center and see if the steam valve on that end has the same amount of opening as the other had. It should and will have the same amount if all adjustments have been carefully made.

46. Adjusting the Governor.—For the purpose of adjustment, block the governor so that the balls stand in the position they would assume at normal speed (about mid-position) and fasten the reach-rod lever m at right angles to a line MN, Fig. 13, midway between the reach rods a and b. Now turn the engine to the point at which cut-off should occur (usually about $\frac{1}{5}$ stroke) and adjust the reach rod for that end, so the valve will trip at that point. The valve and the reach rod for the other end of the cylinder must be adjusted in a like manner. To determine the point of $\frac{1}{5}$ stroke, mark the length of stroke on the cross-head guides and measure off $\frac{1}{5}$ of this from each end. After a few trials, partially rotating the engine back and forth, at the same time making careful adjustments of the reach rods, cut-off can be made to take place at exactly similar points for each end. It is well, now, to lower the governor to the lowest position and observe that the cut-off mechanism does not work, but allows steam to be taken during the full stroke of the piston.

To regulate the sensitiveness of the governor, vary the amount of opening between the two ends of the dashpot cylinder l.

47. Dashpot Adjustments.—Care must be taken, in making adjustments of a valve gear, not to overlook even

the smallest detail. For instance, the dashpot rods must have such a length that the steam arm will be in a position where the hook will surely hook on when the dashpot plunger is home.

48. General Summary.—To regulate the point of cut-off so that the same amount of steam is admitted at both ends, adjust the lengths of the reach rods; to give more or less steam lap, lengthen or shorten the steam rods. A change in the exhaust rods likewise affects the cushion and release. After the eccentric has once been properly set, it is not necessary to disturb it in ordinary cases. If the dashpot rod is too short, the latch will not hook. Look out for this. It is an excellent plan to mark every position; one can then tell at a glance if any adjustments have been disturbed. As a final test of the valve setting an indicator diagram is to be recommended.

CORLISS GEARS WITH TWO ECCENTRICS.

49. In order to secure a satisfactory steam distribution when the steam and exhaust valves of a Corliss engine are both driven by the same eccentric, it is necessary to give the eccentric a certain angle of advance. It has been explained previously that the releasing gear cannot act on the valves after the eccentric has passed its extreme throw position. The result of these conditions is that with a single eccentric the releasing gear cannot be made to act after the piston has passed through a fraction of its stroke always less than $\frac{1}{2}$ and often little more than $\frac{3}{8}$. If the load on the engine is so heavy that cut-off does not take place before this fraction of the stroke has been completed, the steam valves will not be under the control of the governor, but will be closed only by the action of the eccentric.

The range at which cut-off can be controlled by the governor can be considerably extended by the use of two eccentrics and wristplates—one for the steam valves and one for

the exhaust. The exhaust eccentric can then be given the angular advance required by the conditions, while the steam valves may be made to open early enough to secure the desired lead with the steam eccentric set without any angular advance or even at an angle of less than 90° with the crank. The steam eccentric thus reaches its extreme throw position later in the stroke, and the range within which cut-off is controlled by the governor is correspondingly lengthened.

50. Amount to Which the Range of Cut-Off Can Be Economically Extended.—By setting the steam eccentric at a small enough angle with the crank, the range of cut-off with a double eccentric might be extended to nearly or quite full stroke. Such a range, however, is not desirable; it is not economical to use an engine under a load that makes it necessary for steam to follow the piston at boiler pressure for such a large part of the stroke, and, in addition, an attempt to secure a long range of cut-off by setting the eccentric back near to the crank position will cause the steam valves to open very slowly at the beginning of the stroke, thus producing wiredrawing of the steam and reducing the economy when working under normal loads. Satisfactory results can be obtained with the eccentric set at an angle of as little as 81° with the crank, 9° back of the 90° position; this provides for an extreme range of cut-off of about $\frac{7}{10}$ stroke.

51. In designing and adjusting the cut-off gear with a double eccentric motion, it is important to make sure that the steam valve for each end of the cylinder will always be closed by the action of the releasing mechanism before the exhaust valve for that end opens. If this precaution is not observed, there will be danger that with a heavy load the steam and exhaust valves for one end of the cylinder will both be open at the same time, thus allowing steam to blow from the steam chest directly through the cylinder into the exhaust.

SAFETY STOPS.

52. Most, if not all, modern Corliss valve gears are fitted with some kind of a safety stop that will prevent the running away of the engine in case the governor belt should break. In that case the governor will slow down and then stop, and hence, if no safety device is fitted, act as if the engine had slowed down; that is, it will lengthen the cut-off so as to speed up the engine, thus allowing it to run away. To prevent this, some kind of a device is usually fitted that prevents the steam valves from hooking on when the governor stops. When the engine is about to be started, this safety device would prevent the starting of the engine, as the steam valves cannot operate, and hence a collar is usually fitted to the governor by means of which it can be raised sufficiently to allow the steam valves to hook on in starting. As soon as the engine is running this collar obviously must be lowered again so as to render the safety stop operative.

OSCILLATING VALVES WITH POSITIVE-MOTION GEARS.

53. In order to secure the advantages of the four oscillating valves and the wristplate motion of the Corliss gear without the attendant disadvantages of the restricted speed of revolution that is imposed by the releasing gear, and also to secure the positive valve motion and the superior degree of regulation that can be obtained with a shaft governor, a number of engines have been designed with the Corliss valves and wristplate driven by a shifting eccentric controlled by a shaft governor. Since it is often desirable to keep the release and compression constant, some of these engines use two eccentrics. One of the eccentrics drives the steam valves; it is controlled by the governor so as to regulate the amount of valve opening and the point of cut-off in practically the same manner as with a slide-valve engine. The other eccentric is fixed on the shaft and drives the exhaust valves. A variable cut-off with constant release and compression are thus obtained.

54. Engines of this class are often called **high-speed Corliss** engines. The general instructions for centering the wristplate and rocker-arms and for adjusting the valves for lap and lead that were given in Arts. **43** to **45** for the Corliss gear apply to engines with positive-motion oscillating valves. In most cases, however, it will be necessary to give the steam valves an amount of lap considerably in excess of that called for in the table in Art. **42.**

VALVE GEARS.

(PART 2.)

METHODS OF REVERSING ENGINES.

1. Reversing When a Single Eccentric Is Used.—Most engines having valves driven by a single eccentric may have their direction of rotation reversed by merely turning the eccentric on the shaft until its center lies on the opposite side of the shaft and makes the same angle with the crank as before. Such a method may readily be applied to nearly all engines having eccentrics that, like the eccentric of the plain slide-valve engine, are keyed to the shaft or fastened by a setscrew. If the engine is to be reversed with an eccentric keyed to the shaft, it will, of course, be necessary to have two keyways either in the shaft or the eccentric, one for the forward and one for the reverse motion.

2. Reversing Shaft-Governor Engines.—In the case of shaft-governor engines, where the eccentric is directly attached to the governor mechanism, it is generally necessary to shift all parts of the governor to a new position in the governor case, or wheel, to which the governor is attached. For this purpose most governor cases are provided with two sets of holes to which the different parts of the governor may be attached. To reverse the engine, the governor parts must be shifted to the opposite set of holes, after which the weights and springs must be adjusted so as to secure the required speed and degree of regulation. By carefully marking the positions of the weights and

noting the tension of the springs before the governor is taken down and setting it to the same marks in the reversed position, little or no change in adjustment will be required. If the governor case is not provided with a duplicate set of holes, the engine cannot generally be reversed without obtaining a new case.

3. Methods of Shifting the Eccentric When Engine Must Often Be Reversed.—Several devices for easily shifting the eccentric on the shaft are used on engines that must often be reversed, but on which the double eccentric and link motion is for some reason not desirable. These devices are most often used on comparatively cheap engines, such as are employed on traction engines and road rollers. In some cases the eccentric is slotted and attached to the shaft in such a manner that its center may be shifted across the shaft in much the same way as the eccentric of a shaft-governor engine is shifted, the principal difference being that the center of the reversing eccentric may be moved to either side of the shaft so as to run the engine either forwards or backwards, and it is moved by a mechanism operated by hand instead of by a governor. Another method that is sometimes used is to turn the eccentric around the shaft by some form of mechanism; for example, a sleeve carrying a short section of a screw with a large pitch is slid along the shaft; the eccentric is placed on this sleeve and is prevented from moving with it in the direction of the axis of the shaft; it has a thread that fits the screw section of the sleeve, and as the sleeve is moved the screw turns the eccentric to the desired position.

DOUBLE-ECCENTRIC LINK MOTIONS.

4. The most common devices for reversing the direction of rotation of an engine use two eccentrics, one for each direction of rotation; by means of a suitable mechanism, either of these eccentrics may be made to drive the valve at the will of the engineer. The methods that have been most

used for shifting the control of the valve from one eccentric to the other are illustrated by the diagrams, Fig. 1.

5. The **hook,** or **gab gear,** shown in Fig. 1 (*a*), is one of the earliest and simplest forms of the double-eccentric reversing gear. The rods driven by the eccentrics *a* and *b* are provided at their outer ends with the forks or gabs *d* and *e*. As shown in the figure, the gab *d* engages with the valve-stem pin *p*; the valve will, therefore, be driven by the eccentric *a*, and the steam distribution and direction of motion will be the same as though this were the only eccentric. The eccentric rods are suspended by rods *r* and *r'* from an arm of a reversing lever pivoted at *s*. When the reversing lever is shifted so as to bring the outer end *m* to the position *n*, the gab *d* is lifted away from the pin *p* while the gab *e* is raised until it engages the pin. In this position the motion of the valve is controlled by the eccentric *b*, and the direction of motion of the engine will be reversed.

The gab gear, although simple, is not now much used; it has the disadvantage, when compared with the various types of link motion, that with it the point of cut-off cannot be varied except by the use of another eccentric and a separate cut-off valve. Also, it is not well adapted for use with engines that must be reversed when running at high speeds.

6. The **Stephenson link,** Fig. 1 (*b*), is the type of reversing gear now most used. In this gear the eccentrics *a* and *b* are connected to the ends *d* and *e* of a curved link *l*. The pin *p* drives the valve, either through the action of a rocker-arm or by being directly connected to the valve stem, as shown in the figure; this pin is fastened to a block *f*, called the **link block,** which fits in the slot of the link and over which the link slides freely. The link is suspended by the **hanger** *r* from the arm *m* that is pivoted at *s*. By swinging *m* about its pivot *s* the link may be raised or lowered so as to shift it to any desired position relative to the block. As shown in the figure, the link is in the position that brings the pin *p* in line with the eccentric rod *a d*; the motion imparted to the valve will, therefore, be the same as

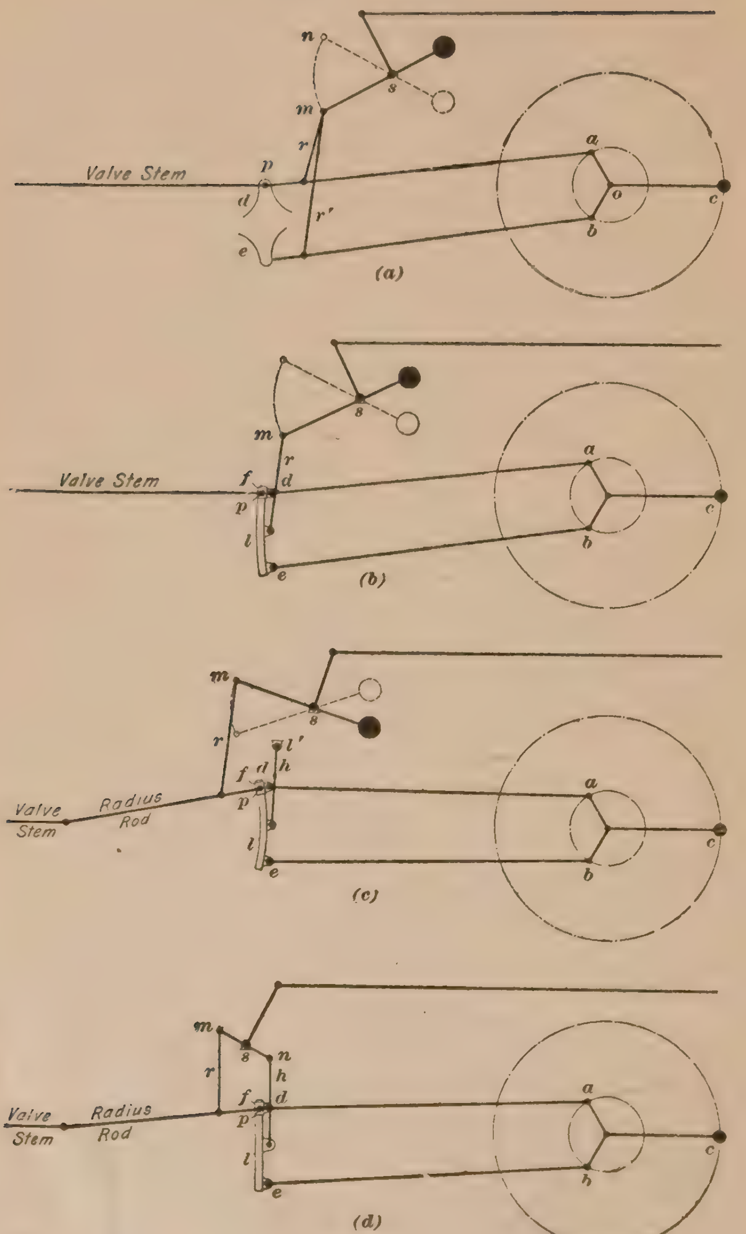

n
s
m
r
Valve Stem
p
d
r'
a
o
c
b
e
(a)
s
m
r
Valve Stem
f
d
p
l
a
c
b
e
(b)
m
s
r
l'
d
h
f
p
Valve
Stem
Radius
Rod
l
a
c
b
e
(c)
m
s
n
r
h
f
d
p
Valve
Stem
Radius
Rod
l
a
c
b
e
(d)

Fig. 1.

though there were only the one eccentric *a*. By lifting the link to its upper position so as to bring the rod *b e* in line with the pin *p*, the eccentric *b* will control the motion of the valve and the direction of rotation of the engine will be reversed.

7. The **Gooch link,** Fig. 1 (*c*), differs from the Stephenson link in that the block is movable up or down in the slot of the link, while the link itself is suspended from a fixed point and cannot be raised or lowered. On account of this fixed suspension, the Gooch link motion is often called a **stationary link motion.** As shown in the figure, *h* is the hanger by means of which the link is suspended from the fixed point *l'*. When the block *f* is raised so as to bring the pin *p* in line with the eccentric rod *a d*, the motion of the valve is controlled entirely by the eccentric *a*; to reverse the engine, the block is lowered so as to bring the pin in line with the rod *b e*.

8. In the **Allen link motion,** Fig. 1 (*d*), the link is straight, and the link as well as the block are moved. To shift the control of the valve from the eccentric *a* to the eccentric *b*, the arm *m n* is turned around the fixed pivot *s* in such a manner that the block *f* is lowered and the link *l* raised until the pin *p* is in line with the rod *b e*.

9. Regulating the Steam Supply With the Link Motion.—The original purpose of the link motion was to reverse the direction of rotation of the engine, but it was soon discovered that by its use the motion of the valve could be so modified as to vary the point of cut-off and so control the amount of work done in the cylinder. When the block is at either end of the link slot, it is evident that the valve will have a range of motion that is determined by the throw of the eccentric connected to that end of the link. Now, suppose the block to be at a point midway between the two ends of the link; an inspection of the figures shows that in this position the block will be acted on equally by both eccentrics, the valve will have only a short range of motion, and the direction of rotation of the engine will not be

determinate. Suppose, however, that the block is at some point between the middle and the upper end of the link slot; the motion of the valve will now be controlled partly by each eccentric, but the eccentric *a* will have more effect on the motion than will the eccentric *b*. Under these conditions the valve travel and the amount of port opening will be less than when the block is at the extreme end of the link, and cut-off will take place earlier.

10. Similarity of the Motion Imparted to the Valve by the Link Motion to That Imparted by the Shifting Eccentric.—By carefully plotting the motion of the valve when driven by the link with the block in different positions or by constructing diagrams of the motion, it is found that the action of the link motion is nearly identical to that of a shifting eccentric whose extreme position corresponds with the positions of the two eccentrics of the link motion. With a shifting eccentric, the lead of the valve for different eccentric positions depends on the path followed by the center of the eccentric as it moves from its extreme to its inner position. With a link motion, as the link block approaches the mid-position from either end and cut-off becomes earlier, the construction of the link motion has the following effect on the lead: With the Gooch or stationary link the lead is constant for all positions of the block in the link, the effect on the lead of shifting the block being practically the same as for a shifting eccentric whose center is shifted along a straight line at right angles to the center line of motion. With the Allen link there is a small variation in the lead as the block nears mid-position, and with the Stephenson link the variation is considerable. With these two the effect on the lead is nearly the same as with a shifting eccentric whose center moves in a curved path.

11. Open and Crossed Rods.—In Fig. 2 (*a*) is shown a diagram of a Stephenson link motion with the link in mid-position. The crank *c* is on its dead center, and the eccentrics *a* and *b* both lie on the side of the shaft next to the link. The eccentric rod from the upper eccentric *a* is connected to

the upper end *d*, and the rod from the lower eccentric *b* is connected to the lower end *e* of the link. When the rods are arranged as here shown they are said to be **open.** In diagram (*b*) of the same figure the eccentrics have the same position on the shaft as in diagram (*a*), that is, when

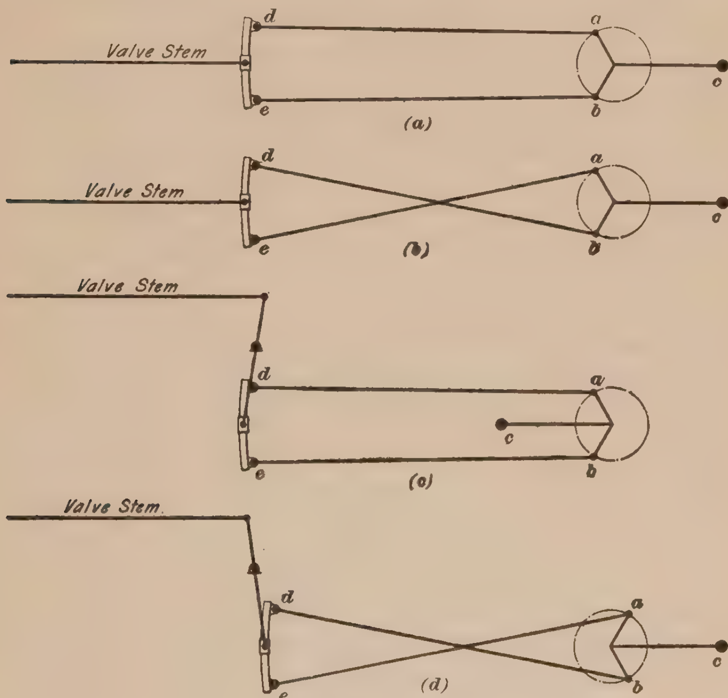


FIG. 2.

the crank is on its dead center they both lie on the side of the shaft **towards** the link; in (*b*), however, the eccentric rod from *a* is connected to the *lower* end *e* and the rod from *b* to the *upper* end *d* of the link, an arrangement that causes the rods to cross each other. When connected up in this manner the rods are said to be **crossed.**

It is important in considering the question of open and crossed rods that the eccentrics be in the position shown in (*a*), (*b*), *and* (*c*), *that is, towards the link, no matter whether the*

crank is on its outer or inner dead center. Thus, suppose the valve to be driven through the medium of a reversing rocker, as shown at (*c*). We know that when a reversing rocker is used the angle between the eccentric and the crank is 90° minus the angle of advance; with the eccentrics lying on the side of the shaft towards the link, this relation between eccentrics and crank will cause the crank to lie on the dead center opposite to that which it occupied in Fig. 2 (*a*), when no rocker-arm was used. This difference in the position of the crank, however, is not to be considered in determining whether or not the rods are crossed. The rods in (*c*) are open the same as in (*a*); although at first sight they appear to be crossed when the crank is on its outer dead center, as at (*d*). A little inspection shows that in this case the eccentrics *do not* point towards the link and that when the shaft is turned they will take the position shown at (*c*); they are, consequently, open.

12. Effect on Lead of Open and Crossed Rods.—As usually made, the radius of the center line of the slot of the link in the Stephenson link motion is equal to the distance from the center of the eccentric to the center of the link block when the link is in its full gear position. With these proportions the effect of moving the link towards its full gear position is to decrease the lead when open rods are used and to increase the lead when crossed rods are used. Conversely, if the link is moved towards mid-gear, the lead *increases* with open rods and *decreases* with crossed rods. With crossed rods the engine can be stopped by placing the link in mid-gear. This cannot be done with open rods, where there is always a small port opening in mid-gear, unless the resistance to be overcome by the engine is so great that enough steam cannot be admitted in mid-gear to run the engine.

13. Setting Valves With Link Motion.—The general method of setting valves with a link motion is to first put the link in full gear position for the direction in which the engine runs the greater portion of the time and set the valve

and the eccentric for this position in the same manner as for the plain slide valve with a single eccentric. The link is then shifted to the opposite full gear position and the eccentric for that position is set. It is to be observed, however, that the lead must be equalized by shortening or lengthening the eccentric rod instead of the valve stem, as was done in the first operation. In following this plan, the result of the first operation of the valve and eccentric setting should be carefully gone over and checked, so as to see that any change that has been made in the later adjustment has not had a disturbing effect on the parts first adjusted.

It often happens that an engine is run for the greater part of the time with the link in the position that will give the most economical point of cut-off for the load to be carried; since the lead with the Stephenson link varies as the position of the link changes, it will generally be found that if the valve is set with the link in full gear position the lead will not be entirely satisfactory for the usual running position. In such a case it will be well to set the valve with the link in the running position.

14. The **Stephenson gear with a straight link** is sometimes used on engines that, like some hoisting engines, are intended to run only with the link in full gear. In such cases the link is intended for use only as a reversing device and never as a means of regulating the steam supply; it serves the same purpose as the gab gear described in Art. **5**, and it has the advantage over that gear of a positive control of the valve during the operation of reversing and of permitting the engine to be readily reversed in any position.

SINGLE-ECCENTRIC LINK MOTION.

15. The **Fink link,** Fig. 3, is a combination of a link with the eccentric strap of a single eccentric in such a way that the direction of motion of the engine or the travel of the valve can be controlled in practically the same manner as with the double-eccentric link motions previously described.

With this link, the eccentric center b is directly in line with the crank radius $o\ c$. The link l is cast solid with one part of the eccentric strap; it is suspended at d by means of a link that swings freely about the fixed point e. As the eccentric revolves, the pin d swings in an arc whose center is e and whose chord is equal to the throw of the eccentric. The up-and-down motion of the eccentric center gives an additional motion to the upper and lower ends of the link, the action

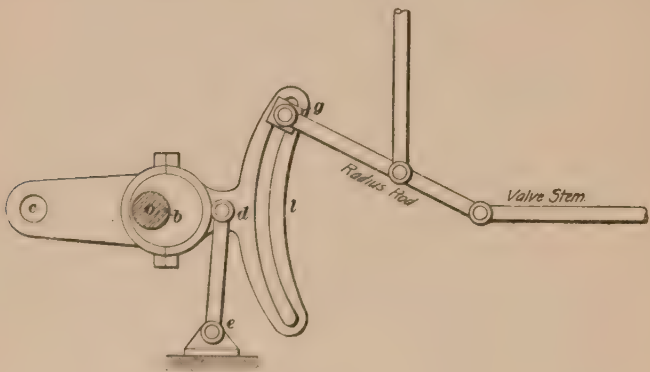


FIG. 3.

being similar to that of a bent lever pivoted at d and having three arms; one of these arms is formed by the eccentric strap proper, which is the arm to which the power is applied; the other arms are the two ends of the link. The combined effect of the two motions is to give the link nearly the same motion as though the two ends were each driven by a separate eccentric. By moving the link block g to different positions in the slot, the direction of rotation of the engine or the point of cut-off can be varied at will.

16. The Porter-Allen Valve Gear.—The Fink link is but little used as a reversing gear, but with the Porter-Allen engine it is used as a variable cut-off gear. Fig. 4 shows the general features of the Porter-Allen valve gear. The eccentric is forged solid with the main shaft and its center c lies on the line of the crank radius and on the same side of the shaft

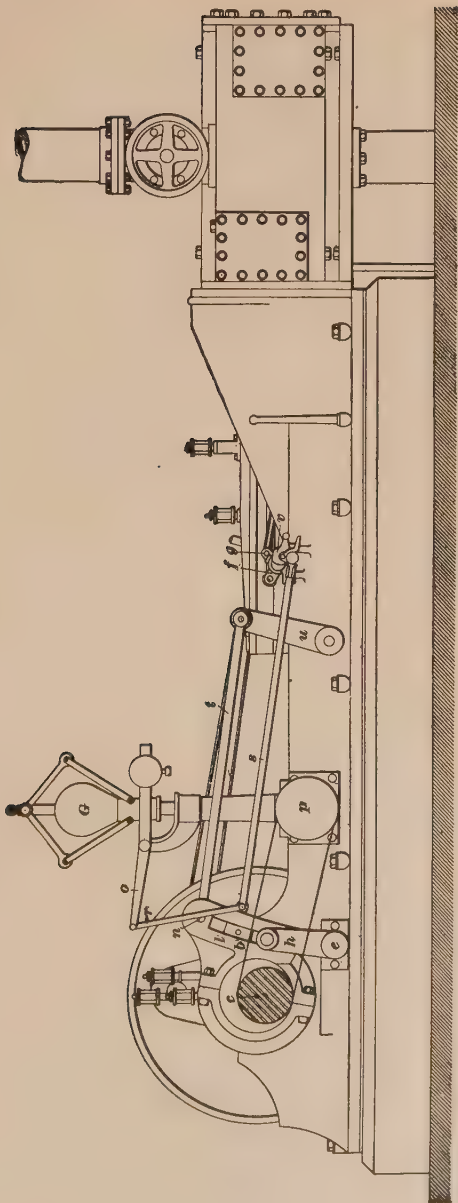


FIG. 4.

center as the crank. The link l is pivoted to the upper end of the swinging arm h, which is pivoted to the engine frame at c. The radius rod s is connected to the link block b; this block is raised or lowered in the link slot by the governor G, acting through the arm o, and the link r, whose lower end is attached to the rod s near the link. The engine has two steam valves, one for each end of the cylinder, and the stem of one of these valves is attached to an arm f and the other to another arm g, both of which are driven by the rod s through the rocker-arm v, the motion imparted to the valves by this rocker being practically the same as though they were driven through the action of the wristplate of a Corliss engine. In addition to the two steam valves there are two exhaust valves that are driven by the rod t pivoted to the upper end of the link at n. The exhaust valves are on the opposite side of the cylinder, and are driven through the rocker-arm u and a rocker-shaft that extends through the engine frame to the other side. The governor is driven by a belt from the shaft to the pulley p. As the load on the engine varies, the governor shifts the block in the link so as to vary the cut-off and thus control the admission of steam to the cylinder. The exhaust valves, being driven from a fixed point on the link, have the same motion for all governor positions; the points of release and compression are, therefore, constant.

Since the link is used only to vary the point of cut-off and not to reverse the engine, the link block need not pass below the point of suspension, and, in consequence, the lower end of the link shown in Fig. 3 is not required.

RADIAL GEARS.

17. There is a type of valve gears known as **radial gears** that perform nearly the same functions as the link motions described in the preceding articles. In these gears the motion imparted to the valve is controlled by a system of arms or links that may have their relative positions varied in such a way that the point of cut-off or the direction of

rotation or both may be readily changed. The principal advantage of these gears is that for some classes of work they give a somewhat more satisfactory steam distribution than can be obtained by the use of a link motion; some of them have the serious disadvantage that they are more complicated than the link motion and therefore more expensive and difficult to keep in good working condition. Radial gears are sometimes used in conjunction with a governor so as to form an automatic cut-off; since, however, they require a considerable amount of force to hold them and to move them to a new position, they are successful only when controlled by a powerful governor, and in most cases the governor action with these gears is too slow to secure good regulation. On account of these difficulties, radial gears are but little used for automatic cut-offs; their principal use is on reversing adjustable cut-off engines in which a special character of steam distribution is desired.

18. The **Joy valve gear,** shown in Fig. 5 as applied to a vertical reversing engine, is a radial valve gear; no eccentric is used, but motion is imparted to the valve by the motion of the connecting-rod *A*. A link *B* is pivoted to the connecting-rod at *a*; its other end is pivoted at *b* to the swinging lever *C*, which is fulcrumed at *c*. At the point *d* on the link *B*, the lever *D* is attached, which is free to turn about the fulcrum *e* at the end of the reversing lever *E*. This lever *E* is free to turn about a pin *f* carried by the reversing rocker shown directly behind the reversing lever *E*. This reversing rocker remains stationary for any running position of the gear, but may, by suitable means, be turned about a fulcrum fixed to the column. With the gear in the position shown, this fulcrum lies directly back of *e*. At *g* the valve-stem connecting-rod *G* is attached. When the engine is in motion the center of the pin *a* will describe the closed curve $a a'$; the center of the pin *d* the curve $d d'$; the lever *C* will oscillate in the arc $b b'$, and with the reversing lever occupying the position $e f$, *g* will describe the curve $g g'$. Assume the crank to be on the top center, as

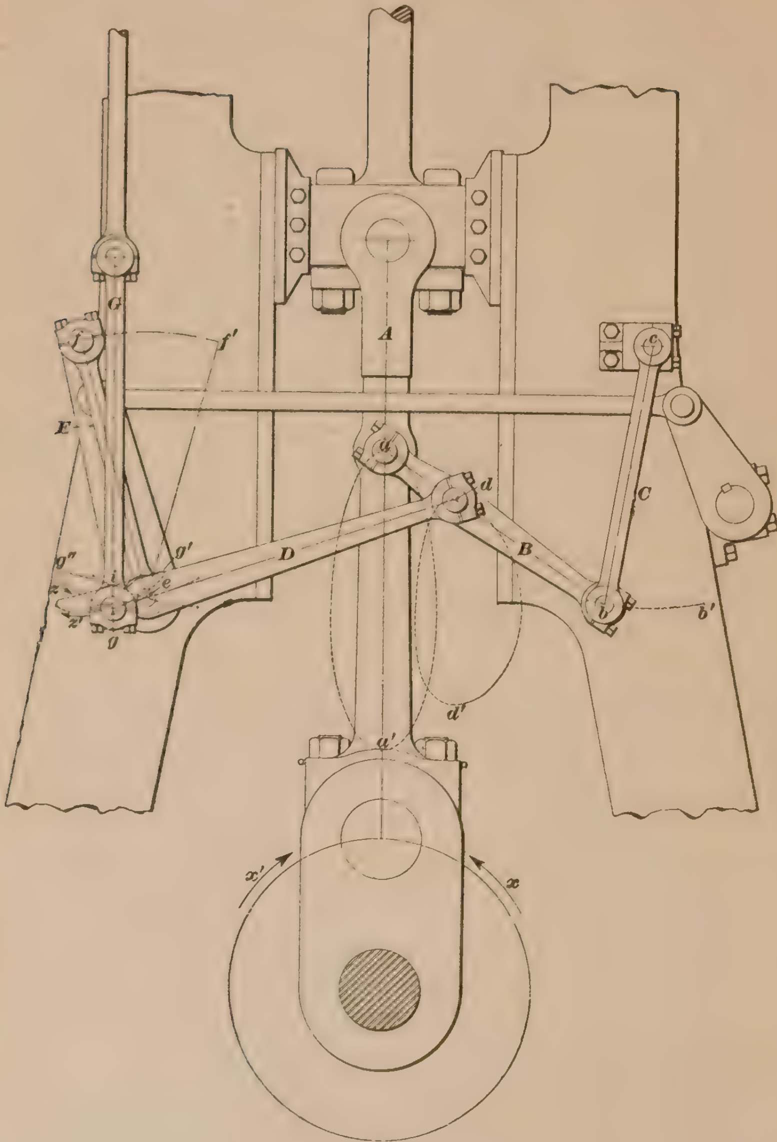


FIG. 5.

shown in the figure. Then the slide valve, which is assumed to be a direct valve, will have opened the upper steam port an amount equal to the lead. If the crank revolves in the direction of the arrow x, the point g moves in the direction of the arrow z', and the valve continues to open the steam port until g reaches the lowest point of the curve $g i g'$. After passing the lowest point, the valve moves upwards, thus closing the steam port. Shortly before the engine reaches the bottom center, i. e., shortly before g occupies the position i, the exhaust port is opened and remains open until g reaches the highest point g'. After passing this point the exhaust port closes quite rapidly, and closes entirely before the crank reaches the top center again. The engine is reversed by shifting the fulcrum f of the reversing lever until it occupies the position f'. The point g will now describe the curve $g i g''$. Assuming the crank to be on the top center, the upper steam port must be opened. Now, since the lowest point of the curve $g i g''$ is to the right of g, it follows that in order that the valve may open the steam port, g must move in the direction of the arrow z; but this cannot take place unless the crank revolves in the direction of the arrow x', that is, in a direction opposite to that in which it revolved when the fulcrum of the reversing lever was at f. The mid-gear position occurs when f occupies a position just midway between the one extreme position in which f is shown and the other extreme position f'. If f occupies any intermediate position, the cut-off will take place earlier than in either extreme position. This gear will give a very early cut-off without unduly increasing the lead and the compression, differing in this respect from the Stephenson link motion, in which, with open rods, as the cut-off occurs earlier, the lead and compression become greater.

Instead of guiding the pivot e of the rod D by means of the swinging lever E attached to a movable arm, the Joy valve gear often uses a slotted link, having in it a sliding block to which the pivot e is attached. By swinging this link around a fixed center, the path of the block and its

pivot e can be varied in practically the same manner as with the construction shown in Fig. 5.

19. The **See-Marshall radial valve gear** is shown in Fig. 6. In this valve gear only one eccentric, shown at A,

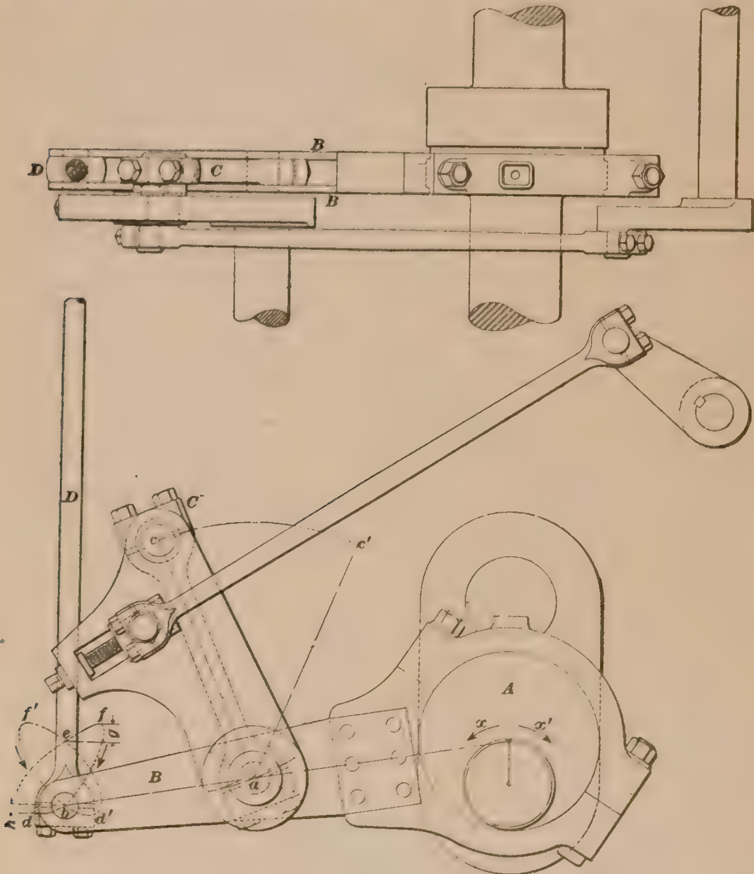


FIG. 6.

is used. It is placed on the shaft so that its center coincides with the center line of the crank. The eccentric rod B is pivoted at a to the radius rod C, which may turn on the fulcrum c. At the free end b of the eccentric rod, the valve-stem connecting-rod D is attached. Assuming the

crank to be on the top center, as in the figure, and the crank to be revolving in the direction of the arrow x, the point b will describe the closed curve $b\,d\,e\,f$, supposing the radius rod to occupy the position $a\,c$. With the crank on the upper dead center, the upper steam port is just uncovered, and the slide valve opens until the lowest point d of the curve is reached. After passing this point the valve closes, the cut-off depending on the amount of lap. When the crank is on the bottom center, the point b is at e, the lower steam port is uncovered and remains open until b reaches the highest point f of the curve, when the valve commences to close. Now, the vertical distance g represents the distance the lower steam port is opened, and h the opening of the upper steam port. A glance at the figure shows g to be greater than h. This involves a later cut-off on the up stroke than on the down stroke — the very thing desired in vertical engines, as the greater power should be developed on the up stroke in order to counteract the weight of the piston, piston rod, crosshead, and connecting-rod. When an indirect valve and a reversing rocker-arm are used, the eccentric will have the same position as is shown in Fig. 6, which is also the correct position for a direct-connected direct valve. When a direct-connected indirect valve is used, and also when a direct valve and reversing rocker-arm are used, the eccentric position will be 180° from that occupied in the figure.

20. The direction in which the engine runs may be reversed by changing the fulcrum c to c'. The crank will now revolve in the direction of the arrow x', and b will describe the curve $b\,d'\,e\,f'$. The cut-off may be readily changed by placing the fulcrum c in any intermediate position, c and c' being its extreme full-gear positions. The nearer c is moved to a point midway between c and c', the less the valve travel will be and the sooner will the cut-off take place. Conversely, the nearer the fulcrum is to c or c', the larger the valve travel and the later the cut-off. In the See-Marshall gear the lead is constant for all cut-offs.

21. Setting the Valve With a Joy or See-Marshall Gear.—Put the gear into the full gear position and place the crank on one center. Shift the valve until the steam port corresponding to the crank position is opened a small amount. Place the crank on the opposite center and measure the lead. Shift the valve one-half the difference between the two leads towards the port where the lead was greatest, in order to equalize the lead. The sum of the two leads is a constant quantity that cannot be changed readily.

POPPET VALVES AND CAM GEARS.

POPPET VALVES.

22. The **Poppet valve** is a type of valve possessing a number of valuable features that has been much used on slow and medium speed engines, but it is not applicable to engines with high rotative speeds. Fig. 7 shows this type of valve as applied to the Putnam engine. The passage *a* connects with the steam chest; *b* is the steam port, and *c* leads to the exhaust pipe. Joining the steam port *b* with each of the passages *a* and *c* are two openings, closed, respectively, by the valves *de* and *d′e′*, each of which consists of two disks with conical edges fitting in conical seats. The disks are rigidly attached to stems *f* and *f′* by means of which they can be lifted from their seats so as to permit steam to pass to and from the steam port or passage *b*. An inspection of the figure shows that in each

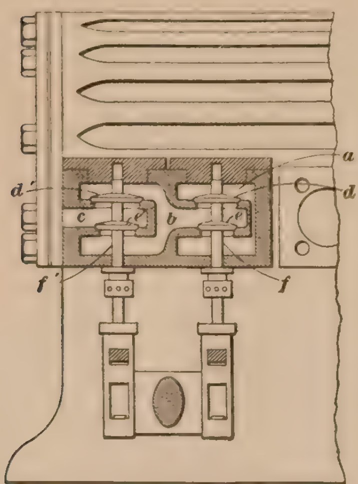


FIG. 7.

case the passage containing steam at the higher pressure connects with the upper surface of the upper disk and the lower surface of the lower disk; by this arrangement the downward pressure on the one is nearly balanced by the upward pressure on the other. The upper disks, however, are somewhat larger than the lower ones, and, in consequence, the downward pressure is greater than the upward. The difference in pressure acts to return the valves promptly to their seats and to firmly hold them there.

23. Single- and Double-Seat Poppet Valves.—An arrangement of two disks on a single stem, so as to control two openings from different directions in such a manner that the pressure on the one is balanced by that on the other, forms what is called a **double-seat poppet valve.** If but one disk with a single opening is used, the arrangement is called a **single-seat poppet valve.**

The double-seat valve can be balanced to any extent desired and can, therefore, be easily operated. It has the disadvantage that great care is required in fitting the two disks so that they will both seat perfectly. Unequal expansion in the valves, the stem, or the seat will make it difficult to keep the valves tight.

BALANCED SINGLE-SEAT VALVES.

24. A method of balancing a single-seated valve that is extremely simple is used in the **Frisbie valve,** shown in Fig. 8, which is a section of the cylinder through the head-end port and valve chambers. The steam valve *a* is a hollow casting of the cross-section shown; the seat is at the lower end. The upper end is turned truly cylindrical to fit a cylinder *c* projecting from the steam-chest cover. Packing rings *b*, *b* insure a tight joint. The lower part of the valve is a little larger than the upper part; owing to the fact that this upper part fits steam-tight in the cylinder, the steam in the steam chest *e* can act only upon the flange. The valve stem passes through a boss central with the outside of the valve and is fastened to it by a nut.

The exhaust valve *d* is slightly different in construction. It is bored out to fit closely a cylindrical projection *f* extending downwards from the exhaust-chest cover. This projection, or piston, is provided with packing rings and is made slightly smaller in diameter than the seat of the valve.

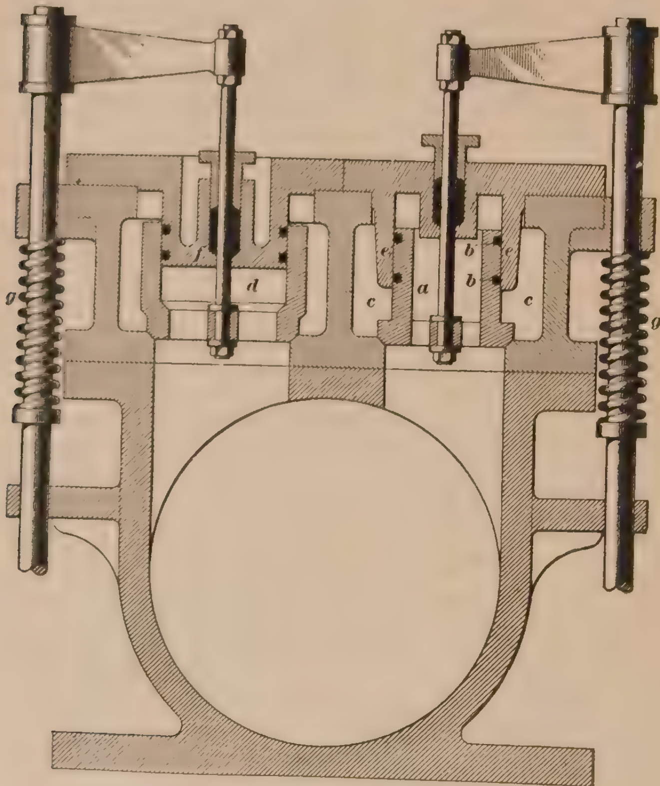


FIG. 8.

Thus, the force acting upon the under side of the valve and tending to open it when live steam is in the cylinder is greatly reduced, the steam acting only upon a ring having an area equal to the difference of areas between those of the seat and piston. The valve is attached to the stem in the same manner as the steam valve.

25. In this design of valve the good qualities of the single-seated poppet valve are retained. With the ordinary double-seated poppet valve the difference in expansion between the seats in the chest and the valve proper makes it quite difficult to get and keep the valve tight, even when ground in while the valve and seats are hot. The ordinary single-seated poppet valve is easy to grind in and keep tight, and this valuable quality is possessed by the Frisbie valve, it being simply a balanced single-seated valve.

CAM GEARS.

26. Cam gears are most often used in conjunction with poppet valves. These gears consist of a set of cams, one for each valve, on a shaft that is usually driven by means of gears from the main shaft, although the cams are often placed on the main shaft itself. The cams are so formed and set on the shaft that each will open and close its valve at the proper time. Since it is possible to give the cam such a form as will enable it to open and close each valve quickly at any desired points in the stroke, the steam distribution can be made almost perfect with this type of gear. At anything above very moderate speeds, however, the sudden striking of the tappets or rollers on the cams makes a great deal of noise and the action of the gearing is unsatisfactory.

27. Variable Cut-Offs With Cam Gears.—A number of different methods have been used for varying the point in the stroke at which with a cam gear the steam valve will be closed. In some cases the steam cam is made with a number of steps, each of which will hold the valve open for a different length of time. By sliding the cam along the shaft the particular step for the required point of cut-off can be made to operate the valve. Tapered cams have also been used which, when slid along the shaft in one direction or the other, held the valve open for longer or shorter periods.

28. The **Putnam automatic cut-off cam gear** is illustrated in Fig. 9. The cam shaft *o* makes but one

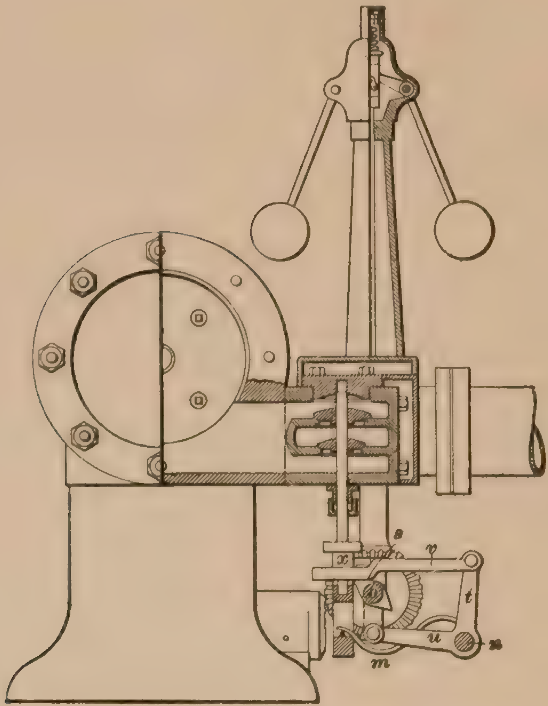


FIG. 9.

revolution to two revolutions of the engine shaft, and the cams have two points; each cam, therefore, opens its valve once during a complete revolution of the engine. As a point of the cam rises by the revolution of the cam shaft towards the right, it strikes against the lever *v* that is pivoted at one end, while the other end fits in a slot in the valve stem *x*. The valve is thus lifted and held open until the point of the cam reaches the shoulder or bend in the lever at *s*; when this point is reached, the lever and with it the stem and valve drop back, thus closing the port. The closing motion of the valve is accomplished partly by gravity, partly

by the difference in pressure on the two disks, and partly by the spring *m*. The point of cut-off is automatically varied by the governor, which moves the bell-crank *u t* pivoted to the shaft *n*, the lever *v* being attached to the arm *t* of the bell-crank. By swinging the lever *t* towards the left the cam point leaves the shoulder *s* on the lever *v* at an earlier point in its revolution, and this permits the valve to cut off the steam earlier in the stroke.

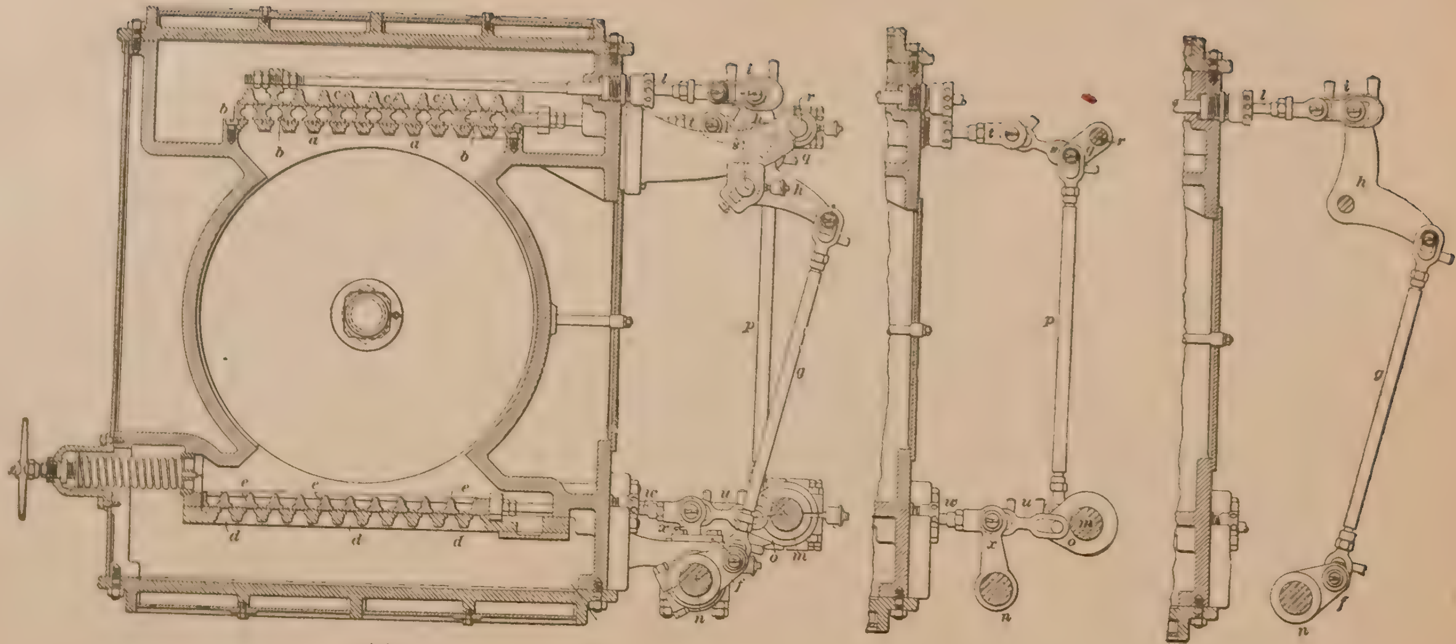
GRIDIRON VALVES: THE McINTOSH & SEYMOUR VALVE GEAR.

29. Gridiron Valves.—Fig. 10 is a cross-section through the ports and valves of one end of the cylinder of a McIntosh & Seymour medium speed engine. The steam-valve seat consists of a number of narrow bars *a*, *a*, with spaces between them for the passage of steam to the port. The steam valve is a frame with a similar series of bars *b*, *b*, *b* having passages between them corresponding to the passages between the bars of the seat. To open the ports, the valve is moved so as to bring the passages between its bars over those between the bars of the seat.

The cut-off valve is made up of a frame with bars *c*, *c*, *c*. It slides on the back of the main steam valve and cuts off the steam supply by closing the passages through the main valve.

The exhaust-valve seat and valve are respectively made up of bars *d*, *d*, *d* and *e*, *e*, *e* similar to the steam seat and valve.

30. Valves of the form here shown are called **gridiron** valves, from the resemblance they have to a gridiron. Their advantage lies in the fact that a liberal port opening can be obtained with a short range of travel; the result is that the wear of the valve and seat and the power required to operate the valve are considerably reduced. The resistance to the flow of steam through the narrow passages between the bars is considerable; the aggregate area of the



(a) (b) (c)

FIG. 10.

passages must, therefore, be greater than the area that would be required with a valve like a plain slide valve that provides for the passage of all the steam through a single large opening, otherwise the steam will be wiredrawn and there will be a loss of pressure in the cylinder. By the use of a sufficiently large number of bars, the aggregate area of opening can easily be made great enough to prevent any serious loss from wiredrawing.

31. The McIntosh & Seymour Valve Gear.—The valves of the McIntosh & Seymour engine are driven by two eccentrics—one, which may be called the main eccentric, for the steam valves and the exhaust valves, and the other for the cut-off valves. The main eccentric is keyed to the shaft, and by means of a short rod it transmits an oscillating motion to an arm keyed to the shaft *m*, Fig. 10, that extends along the side of the engine bed and cylinder. Two cranks, as *o*, are clamped to the shaft *m*, one for each end of the cylinder. Each crank transmits motion to the steam valve and exhaust valve at the end where it is located. The motion of the rocker-shaft *m* is transmitted to the steam valve of the end of the cylinder shown in Fig. 10 (*a*) through the arm *o* and the rod *p* attached to the togglejoint formed by the arm *q* pivoted at *r*, and the link *s* pivoted to the end of the valve stem *t*. To show the relation between these parts more clearly, they have been drawn separately in Fig. 10 (*b*), where they are given the same letters as in Fig. 10 (*a*). A togglejoint connection to the exhaust-valve stem *w* is formed by the arm *o* and the link *u*. The arm *x* is pivoted *loosely* on the shaft *n* and serves as a support or guide for the joint between the end of the valve stem *w* and the link *u*. A similar set of connections serves to drive the steam and exhaust valves for the other end of the cylinder.

32. Effect of Togglejoint on Motion of Valves.—A study of the arrangement of the parts illustrated in Fig. 10 (*b*) shows that for a given angular motion of the shaft *m* the motion imparted to the valves, when the links composing

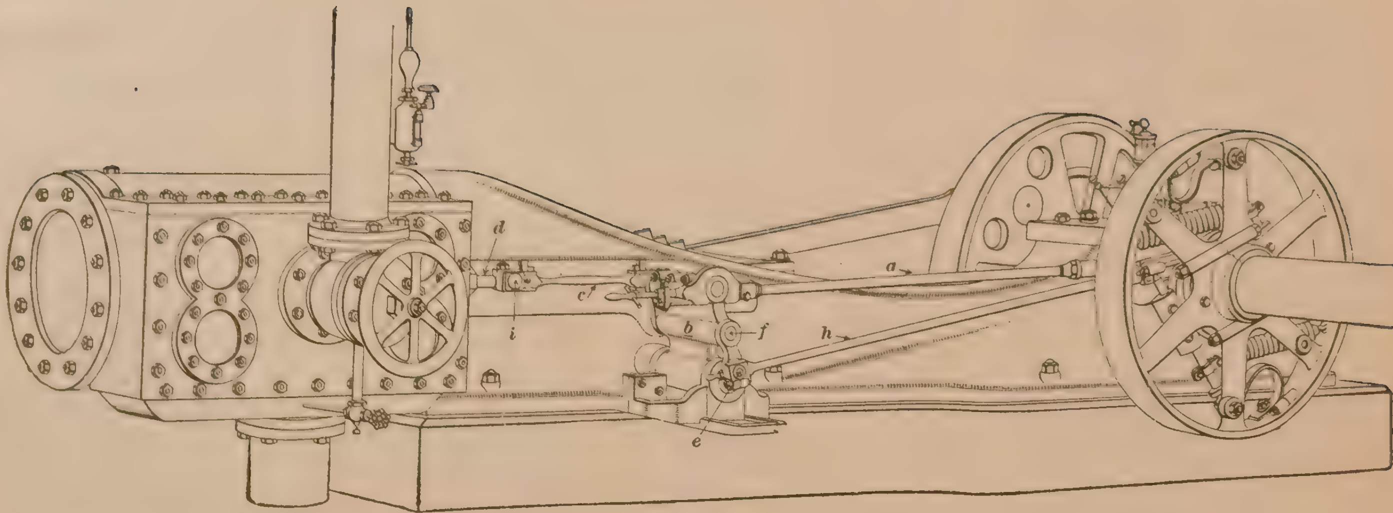


FIG. 11

the parts of a togglejoint lie in nearly the same straight line, is much less than the motion imparted when they form a considerable angle with each other. It is also seen that when one of the toggles, for example the one driving the steam valve, is in the angular position, the parts of the other lie in a straight line. The straight-line position of each of the togglejoints corresponds to the period during which its valve is closed; the valve, therefore, has but little motion during the period of closure. As the joint leaves the straight-line position, the lap of its valve is overcome; the opening motion of the valve takes place while the parts of the toggle are taking and leaving their angular position; the valve thus has a rapid motion during the period of opening and closing, with a period of nearly complete rest during the time when it remains shut. The effect of the toggles is further modified by the action of the eccentric and its connection to the oscillating shaft *m*. By a comparison of the toggle motion with the wristplate motion of the Corliss-engine, it will be seen that the effects on the motion of the valve are quite similar.

33. The **motion of the cut-off valve,** for the end of the cylinder shown in Fig. 10 (*a*), is imparted to it by the cut-off eccentric through connections to an oscillating shaft *n*, the rocker-arm *f*, the rod *g*, the bell-crank *h*, the link *i*, and the valve stem *l*. These parts are also shown by themselves in Fig. 10 (*c*). A similar set of connections transmit motion from the shaft to the other cut-off valve. The cut-off valve is merely a riding valve with fixed edges, and the point in the stroke at which it covers the passages through the main steam valve, and so cuts off the supply of steam to the cylinder, is varied by varying the angle of advance of the eccentric through the action of a shaft governor.

34. Buckeye Valve Gear.—One of the most interesting and successful applications of the riding cut-off valve is that of the **Buckeye valve gear** (see Fig. 11), which shows the gear applied to a Tangye bed engine (see also the sectional view in line with the rocker-arm shown in Fig. 12). In this

gear the main valve is driven by its eccentric through the action of the main eccentric rod *a*, the direct rocker-arm *b*, the link *c*, and the valve stem *d*. As will be seen, the main valve is driven in exactly the same manner as a simple slide valve with a direct rocker.

The cut-off valve eccentric rod *h* is attached to the pin *e* of one arm of a rocker that is pivoted in the main rocker *b* by the rocker shaft *f*. The other arm of this rocker carries the pin *g*, Fig. 12, to which the cut-off valve stem is attached. The link *c* of the connection to the main valve is attached to the pin *k* of the main rocker. As will be seen by reference to Fig. 12, the center line of this pin, and consequently

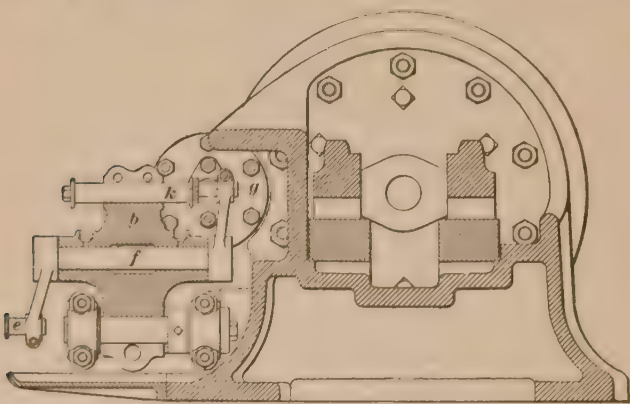


FIG. 12.

of the link *c*, is at one side of the center line of the valve stem stuffingbox, while the pin *g* is directly in this latter line. The reason for this is that the cut-off valve works inside of the main valve, which is either a hollow cylinder or a special hollow box, and its valve stem leads to the inside of the main valve through the center of the hollow main-valve stem. The main-valve stem is driven by the link *c* through a clamp carrying the pin *i*, Fig. 11.

35. The **Buckeye governor** varies the point of cut-off by changing the angle of advance of the cut-off eccentric; the throw of the cut-off valve is, therefore, constant for any

governor position. The effect of this constant throw in conjunction with the combination of the cut-off rocker-arm with that of the main valve is to give the cut-off valve a constant range of travel over the main valve. This keeps the wear of the cut-off valve seat constant throughout its whole length and prevents the formation of shoulders or inequalities in the surface of the seat that would result in causing the valve to leak.

36. The **Buckeye valve,** Fig. 13, is a hollow box having two openings through the top that connect with similar

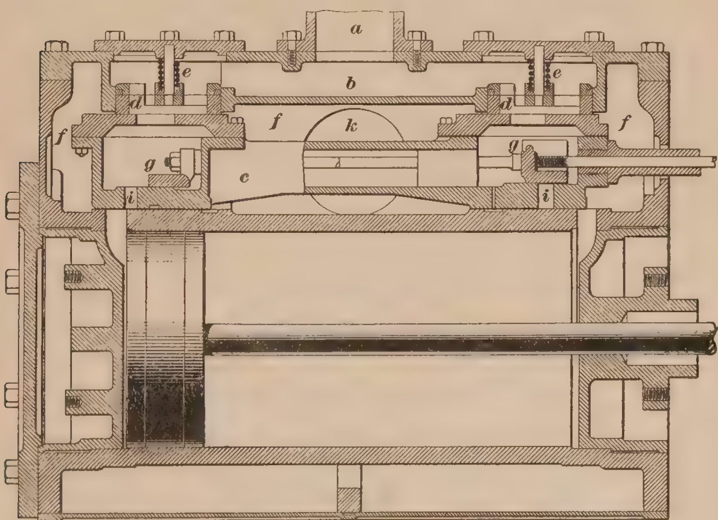


FIG. 13.

openings in a chamber formed in the steam-chest cover. Live steam enters the steam chest through the steam pipe *a* and flows through the passages *b*, *b* into the interior of the main valve *c*. The main valve is held to its seat by the balance pistons *d*, *d* and the coiled springs *e*, *e*; these balance pistons rest on top of the main valve, and while allowing it to slide back and forth, prevent live steam from entering the exhaust cavity *f* of the steam chest. The main valve has ports *i*, *i* through which live steam is admitted to

the cylinder when these ports are in line with the steam ports. Steam is exhausted past the outside edges of the main valve, which is hence an indirect valve. Valve seats are formed inside of the main valve on which the plates *g*, *g* of the cut-off valve work. Cut-off is effected by the cut-off valve closing the port *i* in the main valve. The exhaust steam leaves the port *f* of the steam chest through the exhaust pipe *k*.

37. Setting the Valves of the Buckeye Gear.—The main valve is set in exactly the same manner as any ordinary slide valve, shortening or lengthening the main-valve eccentric rod to equalize the lead and shifting the eccentric to obtain the desired amount of lead. The eccentric having been fastened, block the governor weights about half way between their inner and outer positions. Turn the engine over by hand and note at what part of the forward and backward stroke cut-off takes place. If the cut-off is unequal, change the length of the rod leading from the governor eccentric to the rocker-arm in a direction that will move the cut-off valve towards the end where cut-off is later by an amount equal to one-half the difference in the cut-offs. The valves are now set.

HINTS FOR ADJUSTMENT AND CARE OF VALVE GEARS.

38. Most Designs Covered by a Few General Principles.—Nearly every principle involved in the construction of the governors and valve gears now used in stationary practice is illustrated and explained in connection with the description of the types that have been described in the preceding articles. These principles have been applied in a great variety of ways, and in some cases they have been so modified and obscured by the peculiar form of the different parts of the mechanism as to be scarcely discernible. A little careful study, however, will generally reveal the true character of the device and enable the engineer who has the requisite

skill and experience in the use of tools to apply his knowledge of the general principles so as to make the adjustments required to keep the engine in satisfactory running order.

39. Study of Relation of Different Parts to One Another.—Before trying to make adjustments of any kind, it is of the utmost importance to know the purpose of each part and the effect that will be produced by a change in its adjustment. By keeping this principle in mind much trouble will be prevented and the danger of broken or damaged connections will be greatly reduced. If, after a careful study of the machine, there is any doubt as to the purpose of any part, or if the means for making adjustments that seem to be needed cannot be discovered, it is best to apply to the makers for instructions. Since most engine builders make changes from time to time in the details of their machines, it is always best when asking for instructions or information to state clearly the size and type of engine and, if possible, to give its number and the date at which it was built and installed.

40. Keep All Joints Carefully Adjusted.—All lost motion in the various connections of a valve gear has a disturbing effect on the action of the valve; with the numerous joints existing in some gears the aggregate effect of a small amount of slack in each will be considerable and may seriously affect the steam distribution. The noise produced by the pounding and rattling of loose joints is also very annoying and the rate of wear increases rapidly with the increase in the amount of lost motion. It must, however, be remembered that joints may be too tight as well as too loose, and that the effect if the joint is too tight will be more speedily and surely disastrous than it will be if there is a moderate amount of lost motion.

41. Two types of joints, each of which has its special advantages, are used in valve gears. The first type is the solid, or non-adjustable, joint. It consists merely of a pin fitted in a hole or eye in the link. If a joint of this kind becomes so much worn as to give trouble, the eye may be bored out and a bushing that has been carefully fitted to

the pin forced into it. In many cases the eye is originally fitted with a bushing that can be readily renewed when worn. A joint of this type requires but little attention, beyond being well oiled when in use, and it cannot be injured by ignorant and careless attempts at adjustment. In the hands of a skilled and careful man, however the second, or adjustable, joint, if properly designed, is generally preferred. With it all wear and lost motion can be taken up as fast as may be desirable, and the gear can thus be kept in good working order until the joint is worn out.

With adjustable joints it is important to note the effects of "keying up" or tightening the joints on the length of their rods. If the change in length has an undesirable effect on the motion, it may be prevented or compensated for by the use of liners placed back of the half of the bearing that is not affected by the key or setscrew.

Adjustable joints must be carefully watched to prevent the working loose of bolts, nuts, and cotters and of the jam nuts, setscrews, or other locking devices by which they are held in place. Attention to these details will prevent much trouble.

42. Setting Valves for Equal Leads and Cut-Offs.—On account of the disturbing effect of the angularity of the connecting-rod, it is generally impossible to so set the valves of an automatic cut-off engine that cut-off for different loads will be equal for both ends of the cylinder. In some gears, however, the effect on the point of cut-off of the angularity of the rod is neutralized by giving the eccentric rod an angle with the rocker that will affect the valve motion in the same way that the angle of the connecting-rod affects the motion of the piston; with such a gear, the valve may be set for equal leads, and equal cut-off will be obtained for the whole range of cut-off.

43. When no provision for equalizing the cut-off is made, it is usual to set the valve for equal leads and ignore the irregularity in cut-off. In some cases, however, the behavior of the engine can be improved by departing from this rule. In vertical engines it is well to give the valve more lead for

the lower port than is given for the upper one; this secures an earlier admission of steam to the lower end of the cylinder and aids in arresting the downward motion of the reciprocating parts. In many cases of fixed cut-off engines it will be found that quiet and smooth running will be more easily secured if the valve is set to give points of cut-off more nearly equal than would be obtained by setting for equal leads. With automatic cut-off engines that work under a nearly uniform load, or a load that varies above and below a certain value that obtains for the greater portion of the time, it will often be desirable to set the valve for nearly equal cut-offs at the usual working load.

44. Tests for Valve Setting.—One of the most useful indications of a satisfactory setting of the valves is a quiet- and cool-running engine. If the crankpin and bearings, when carefully lined, adjusted, and oiled, persistently get hot and noisy, it is a strong indication that the setting of the valves is unsatisfactory for the conditions of the load, speed, and steam pressure under which the engine runs. What changes are needed to secure more satisfactory results can best be determined by a little experiment, guided by experience and judgment. An indicator diagram will, however, serve a very useful purpose as an indication of the changes that will most likely be beneficial, and such diagrams should be taken whenever possible. Useful as the indicator is for this purpose, it is well to remember that the particular setting of the valves that will give a fine diagram will not necessarily give the best results as far as quiet and cool running is concerned.

45. Pressure Plates and Packing Rings.—Pressure-plate valves demand much careful attention to prevent leakage and a consequent waste of steam. The plate, valve seat, and valve must be carefully fitted; the valve must work freely between the plate and seat without being loose enough to permit of the passage of any considerable amount of steam. If the plate is adjustable by means of wedges or screws, it must be let down on the valve with great care; if

it rests on edges that must be planed or scraped, the difficulties are still greater and the adjustment requires a considerable degree of skill. The only adjustments possible with piston valves are those involved in setting the rings so that they will fit their seats without working too tight. With both pressure-plate and piston valves, the engine should be warmed gradually by admitting steam slowly before starting up, otherwise the unequal expansion of the valve and its surroundings may cause the valve to stick and break the valve gear.

46. Packing and Lubrication.—Valve stems should be packed so as to be only just tight enough to prevent an annoying leakage of steam. This is especially true in the case of automatic cut-off engines, where any friction of the valve and valve stem has a disturbing effect on the governor action. On account of the effects of friction on the governor, as well as to prevent wear and loss of power, it is also highly important that all parts of the gear, from the valve and seat to the last joint in the governor, be thoroughly and uniformly lubricated. This condition is not met by periodically flooding the joints with oil; efficient lubrication can be secured only when the supply of oil is uniform and steady, the best method of lubrication being that in which the parts run continually in an oil bath.

47. Marks for Determining Positions of Valves and Connections.—The use of marks for obtaining the relative positions of the edges of the valves and ports and for setting the wristplate of Corliss engines was explained in the articles on setting Corliss valves. In nearly every type of gear it will be found that somewhat similar marks can be made that will be of great assistance in making adjustments and for determining whether or not the adjustments have been disturbed. With a slide valve that is not adjustable along the stem, the positions in which the valve just covers one or the other of the steam ports, and, in consequence, the edges of valve and port are in the same line, can be located by a tram and center-punch marks on the valve stem and some convenient point on a fixed part of the engine bed, steam

chest, or cylinder. These marks may be conveniently established in the following manner: First, select a fixed point in some convenient place—the end of the steam chest, as shown by the point *a*, Fig. 14, will generally serve well for this purpose—and mark it plainly with a center punch. Prepare a tram (see *b*, Fig. 14) of such a length that when the point on one end rests in the center-punch mark just made, the other will reach to a convenient point on the valve stem.

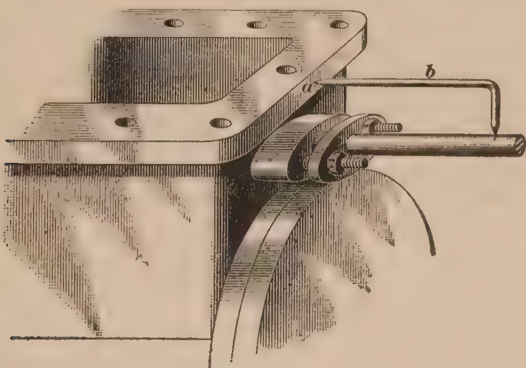


FIG. 14.

With the steam-chest cover off, set the valve so that it just covers one of the ports and make a center-punch mark on the valve stem at the point to which the tram reaches when one end is in the fixed center-punch mark, as shown in the figure. Shift the valve so that it will just cover the opposite port and make a similar center-punch mark on the stem for this position. With the tram and these marks, the positions of the valve where it just covers each port can be determined and the valve can be set without the necessity of removing the steam-chest cover.

48. Locating the Port Opening.—With some valves the point in the valve travel at which the ports begin to open cannot be easily determined by direct observation, owing to the fact that the edges are more or less hidden by the form of the valve or its seat. If the engine can be connected with a steam supply, the points where port opening

begins can be located in the following manner: Open the drain cocks and disconnect the eccentric rod from the rocker or valve stem. Put the crank on either center and block it there. Turn steam into the steam chest and move the valve by hand until steam just begins to blow through the drain cocks, first at one end of the cylinder and then at the other The valve position at which steam begins to blow from the cock at either end of the cylinder will be the position at which the port for that end begins to open. Make center-punch marks on the valve stem for these positions of the valve to suit a tram, and a record will be had from which the valve can be set at any time. If the valve is not well fitted and leaks badly, it will be difficult to determine by the above method the point at which the edges of valve and port are in line; in most cases, however, the plan will be found to give good results.

49. Locating the Port Opening by Indirect Measurement.—It is sometimes impossible or inconvenient to determine by direct observation the points at which a valve begins to open the ports. In this case the points may be located by indirect measurements in the following manner: Let Fig. 15 represent a section of a cylinder and steam chest with a direct valve. After removing the valve, select some point in the steam chest that will not be covered by the valve when it is in position, and from which measurements can be easily taken to the steam ports and to some well-defined surface of the valve when the latter is in

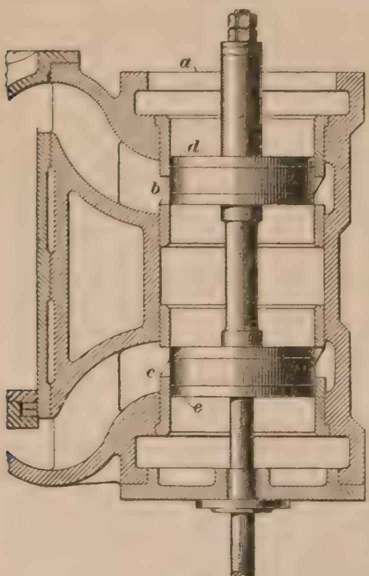


FIG. 15.

place. Suppose the point selected to be in the plane of the surface a. Measure the distances $a\,b$ and $a\,c$ from the surface a to the cut-off edges of the steam ports. Also measure the distance $d\,e$ between the cut-off edges of the valve. Evidently the cut-off edge d will be in line with the cut-off edge b when the distance $a\,d = a\,b$. Likewise, the cut-off edges c and e will be in line when the distance $a\,d = a\,c - d\,e$.

50. In a direct valve the measurement from the point of the steam chest that has been selected as most convenient can generally be made directly to the nearest cut-off edge of the valve. In an indirect valve, however, the cut-off edges are hidden, and hence the measurements must be made to some well-defined surface of the valve that can be reached, taking the distance of this surface from the cut-off edges into account.

In Fig. 16 an indirect valve is shown, where b and c are the cut-off edges of the ports and d and e the cut-off edges

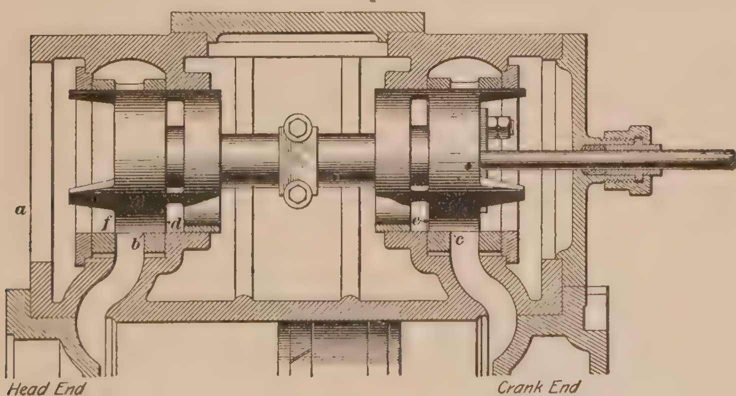


FIG. 16.

of the valve. The surface a of the steam chest and the surface f of the valve are the most available surfaces to measure from. Reflection will show that b and d will be in line when the distance $a\,f = a\,b - d\,f$. Likewise, the edges c and e will be in line when the distance $a\,f = a\,c - e\,f$.

51. Setting an Indirect Valve Attached to a Shaft Governor.—Fig. 16 illustrates an application of the method of indirect measurements to the setting of the piston valve of a center-crank, high-speed McIntosh & Seymour engine. In the illustration the steam laps are the distances *b d* and *e c*; there is no exhaust lap.

To set the valve, remove the bonnet on one end of the steam chest and remove the valve. From some well-defined face of the valve, as *f*, measure the distances *f d* and *f e*. From some well-defined face of the steam chest, as the face *a*, measure the distances *a b* and *a c* to the edges of the ports past which steam is admitted, which are the edges *b* and *c* in this case. Replace the valve, put the crank on the head-end dead center, and block the governor about half way open. Since in nearly all shaft-governor engines the lead is a fixed quantity that cannot be readily changed, it is seen that in valve setting nothing is to be done except to set the valve so that the lead is equal on both ends.

Lengthen or shorten either valve stem or eccentric rod until the distance from the face *a* to the face *f* is equal to $a\,b - (d\,f +$ the amount of lead it is believed the eccentric has been set for). Thus, assuming $a\,b = 5$ inches, $a\,c = 19$ inches, $f\,d = 2\frac{1}{4}$ inches, and $f\,e = 14\frac{1}{4}$ inches, and that it is believed $\frac{1}{4}$ inch is about the right lead, move the valve until its face *f* is $5 - (2\frac{1}{4} + \frac{1}{4}) = 2\frac{1}{2}$ inches from the face *a*. Place the crank on the opposite dead center and measure *a f*. If this distance is more than $a\,c - (f\,e -$ the assumed lead), the valve stem or eccentric rod must be lengthened. Thus, if the distance *a f* is $5\frac{1}{4}$ inches and $a\,c - (f\,e -$ assumed lead) $= 19 - (14\frac{1}{4} - \frac{1}{4}) = 5$ inches, it shows that the valve must be moved towards the head end of the cylinder by one-half the difference between *a f* and $a\,c - (f\,e -$ assumed lead), or $\frac{5\frac{1}{4} - 5}{2} = \frac{1}{8}$ inch. Conversely, if this distance *a f* is less than $a\,c - (f\,e -$ assumed lead), the valve must be moved towards the crank end by shortening the valve stem or eccentric rod by one-half the difference. Thus, assume *a f* to be $4\frac{3}{4}$ inches, while $a\,c - (f\,e -$ assumed lead)

$= 5$ inches. Then, shorten the valve stem or eccentric rod $\frac{5 - 4\frac{3}{4}}{2} = \frac{1}{8}$ inch. The valve is now correctly set for equal lead.

52. If desired, the setting may be checked as follows: the crank being on the crank-end dead center, measure the distance $a\,f$. Add the distance $f\,e$ to it and subtract from it the distance $a\,c$. The remainder will be the lead at the crank end. Now place the crank on the head-end dead center and measure $a\,f$ again. If $a\,f$ is equal to $a\,b - (f\,d +$ lead at crank end), the valve is correctly set for equal leads. If this is not the case, it shows that the position of the valve was not changed the correct amount.

53. Setting a Plain Indirect Valve.—When an indirect piston valve is not controlled by a shaft governor, the lead of the valve depends on the angular advance of the eccentric, and, consequently, can be readily changed. To set such a valve, take the measurements $a\,b$, $a\,c$, $f\,d$, and $f\,e$ first of all. Put the crank on the head-end dead center after the valve has been replaced and connected up, and set the eccentric, as near as can be judged by eye, where it ought to be. Lengthen or shorten the valve stem or eccentric rod until the distance $a\,f = a\,b - d\,f$. Place the crank on the opposite dead center and measure $a\,f$. If this distance is more than $a\,c - f\,e$, the valve must be moved towards the head end one-half the difference between $a\,f$ and $a\,c - f\,e$. Conversely, if $a\,f$ is less than $a\,c - f\,e$, the valve must be moved towards the crank end one-half the difference between $a\,f$ and $a\,c - f\,e$. This adjustment having been made, place the crank on the crank-end dead center and shift the eccentric until $a\,f = (a\,c +$ desired lead$) - f\,e$, or place the crank on the head-end dead center and shift the eccentric until $a\,f = a\,b - (f\,d +$ desired lead). Fasten the eccentric. This will complete the valve setting, which will be for equal leads. To check the setting, proceed exactly as explained in the previous article.

54. Verify All Adjustments.—When setting valves or adjusting the gearing in any way, the work should be carefully verified before it is finally regarded as finished. It very often happens that a change in one part will affect another that has previously been adjusted, and if the work is not carefully examined before it is left, the effect of the change will not be discovered until it is found that the engine does not work well.

After making any changes or adjustments the engine should be turned over slowly, by hand if practicable, and the motion of each part should be carefully watched in order to make sure that there are no interferences and that all parts work freely.

Mark the parts after the adjustments have been verified so that they can be put back to the same place if they are disturbed in any way.

CONDENSERS.

INTRODUCTION.

EFFICIENCY.

1. The perfect combustion of 1 pound of good coal will produce 14,000 B. T. U. and 1 B. T. U. is equal to 778 foot-pounds.

The mechanical energy stored up in 1 pound of good coal is, therefore, $14,000 \times 778 = 10,892,000$ foot-pounds. Reducing this to horsepower per hour, we have $\frac{10,892,000}{33,000 \times 60} = 5.5$ horsepower, nearly. This means that if all the heat made available by the burning of a pound of coal was transformed into work, it could be done at the rate of 5.5 horsepower per hour.

2. We know from practical experience that 1 pound of coal never has produced 5.5 horsepower per hour, about the best record ever having been made giving 1 horsepower per hour for 1 pound of coal. This comparison shows what an imperfect apparatus a steam engine really is when considered as a heat engine.

3. We will now endeavor to trace out what becomes of all this heat that is produced by the combustion of the fuel.

In the first place, a very considerable part of the heat produced by the combustion of the fuel escapes in various ways from the boiler, the largest part of which passes up the stack with the gases of combustion. All this heat is not actually wasted, however, as a large part of it is necessary to produce draft, but it is lost so far as doing work in the cylinder is concerned. Then, the losses by radiation, blowing off with the blow-off valves, the safety valve, and the gauge-cocks, and various other small losses make up the total loss from the boiler.

This loss varies greatly in different boilers, but a fair average may be taken at 4,000 B. T. U. This leaves 10,000 B. T. U. in the steam that goes to the engine and represents the part of the heat available for doing work in the cylinder.

4. As each unit of heat in the steam is capable of doing a definite amount of work, it is obvious that the more heat that is available for this purpose, the greater the amount of work can be performed, and the doing of this work by the steam is accompanied by a reduction of temperature. From this it is easily seen that the amount of work which can be done by steam depends on the decrease in temperature.

5. According to the now universally accepted theory, heat energy consists of the motions of the molecules of the hot body. In order, therefore, to change into work *all* the heat energy contained in the steam, it will be necessary to take from the steam all its molecular motion; in other words, to lower its temperature until its molecules will be at rest.

From experiments it has been concluded that the temperature at which a body will be in such a state is about 460° below zero, Fahrenheit. This point is called the **absolute zero** of temperature. Temperatures measured from this zero point are called **absolute temperatures.**

Absolute temperatures are obtained by adding 460° to the ordinary temperature. That is,

Absolute temperature = ordinary temperature + 460°.

For example, the ordinary temperature of steam at atmospheric pressure is 212°. The absolute temperature is 460 + 212 = 672°. This means that if steam or other gas at 212° could be cooled down 672°, the molecules would cease moving and there would be no heat in the body.

It is at once evident that it is impossible practically to cool the steam leaving the engine cylinder to even approximately so low a temperature; long before reaching the absolute zero, the steam would be changed to ice. This explains why it is impossible for an engine to obtain 778 foot-pounds from each B. T. U. conveyed to it.

6. Let T_1 denote the absolute temperature of the steam entering the engine cylinder and let T_2 represent the absolute temperature of the steam leaving the cylinder. If we take the amount of energy contained in the entering steam as proportional to the absolute temperature T_1, then it may be proved that the amount of work extracted by the engine is proportional to $T_1 - T_2$; or,

$$\text{Useful work} : \text{total energy} :: T_1 - T_2 : T_1.$$

The efficiency of an engine is the ratio of the useful work to the total energy; therefore,

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1}.$$

The above reasoning may perhaps be made clearer by comparing the temperature of steam to the "head" of a water-power. If a source of water is 1,500 feet above the sea level, the work or potential energy of the water is represented by the head of 1,500 feet. But possibly the water-wheel to which the water is led is itself 1,400 feet or more above the sea level; in this case, only 100 feet of the 1,500 is available head, and the efficiency of the arrangement

cannot exceed $\frac{1,500 - 1,400}{1,500} = \frac{1}{15}$. Absolute zero is the "sea level" of temperature, but as far as steam is concerned, it is utterly impossible to lower its temperature to the absolute zero by reason of its changing into ice at 32° F., corresponding to 492 absolute. About the lowest temperature to which the steam can be lowered in practice is 562° absolute. This temperature corresponds to an absolute pressure of 1 pound, which is probably the lowest pressure attainable.

The above expression for the efficiency applies equally well to steam engines, hot-air engines, gas engines, or any other form of heat engine.

CONDENSATION.

7. It is plain that the only two ways of increasing the efficiency of the steam engine is either to raise the temperature T_1 of the live steam or to lower the temperature T_2 of the exhaust; T_1 may be raised by increasing the boiler pressure; T_2 may be lowered by using a **condenser.**

In non-condensing engines, that is, engines that are not supplied with a condenser, the steam is exhausted into the atmosphere, and therefore the exhaust steam must have, at least, the pressure of the atmosphere; in practice, the back pressure of steam in a non-condensing engine is scarcely ever less than 16 pounds above vacuum, and is oftener 17 pounds or more. In good condensing engines the back pressure is often as low as 2 pounds above vacuum.

Suppose the boiler pressure of the steam is 80 pounds absolute (above vacuum); the temperature corresponding to the pressure is, from the Steam Table, 311.9° F., and the absolute temperature is, therefore, 460° + 311.9° = 771.9° F. The absolute temperature corresponding to a pressure of 17 pounds is 460° + 219.5° = 679.5° F., and corresponding to a pressure of 3 pounds is 460° + 141.7 = 601.7° F. The thermal efficiency of the engine, if non-condensing, is $\frac{T_1 - T_2}{T_1}$

$= \frac{771.9 - 679.5}{771.9} = 12$ per cent., nearly; if condensing to 3 pounds (absolute), the efficiency is $\frac{T_1 - T_2}{T_1} = \frac{771.9 - 601.7}{771.9}$ $= 22$ per cent.

8. The increase of economy by the use of the condenser may be shown in another manner. Let *A B C D E F*, Fig. 1, be an indicator diagram from a non-condensing

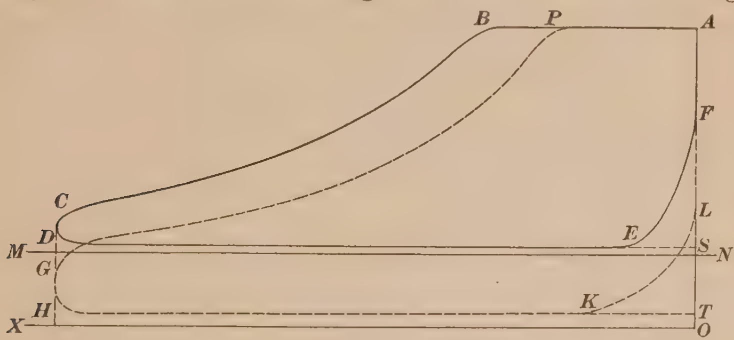


FIG. 1.

engine. *M N* is the atmospheric line and *O X* the vacuum line. The back pressure, as shown by the diagram, is *O S*. The area of the diagram represents, to some scale, the work done per stroke. Now let a condenser be attached to the engine. The back pressure will be lowered to *O T*, the line *H K*, instead of *D E*, now being the lower line of the diagram, and *A B C H K L* will be the new diagram, its area, as before, representing the work done per stroke. Hence, by adding a condenser to the engine, the work per stroke has been increased by an amount represented by the area *F E D H K L*, the steam consumption remaining the same. Suppose the steam to be cut off at a point *P*, making the area of the diagram *A P G H K L* equal to the area of the original diagram *A B C D E F*. Then, the work per stroke is the same in both engines, but the condensing engine uses an amount of steam per stroke represented by the length *A P*, while the non-condensing engine uses an amount represented by *A B*. Either case shows the economy of the condenser.

THEORY OF THE CONDENSER.

FORMATION OF VACUUM.

9. If a cubic inch of water is converted into steam at the atmospheric pressure, it will occupy 1,646 cubic inches of space, and, conversely, if 1,646 cubic inches of steam at the atmospheric pressure are condensed into water, it will occupy but 1 cubic inch of space; hence, if a closed vessel is filled with steam at the atmospheric pressure and that steam is condensed to a cubic inch of water, $\frac{1645}{1646}$ of the space will be theoretically devoid of air or any other known substance and a perfect vacuum would be the result. This is not strictly true in practice, however, from the fact that the feedwater of the boilers always contains a small quantity of air, which passes into the condenser with the exhaust steam and is released there when the steam is condensed; more or less air also finds its way into the condenser through leaks around the piston rod and valve stems, and in the case of the jet condenser and the induction condenser, the air contained in the condensing water is also released in the condenser under the influence of the partial vacuum. There is still another obstacle in the way of producing a perfect vacuum in the condenser, which may be explained as follows: Water in a vacuum emits a certain amount of vapor, and vapor is also formed in a jet condenser and in an induction condenser by the heat in the exhaust steam being imparted to the condensing water; therefore, if the condenser were successively filled with steam and the steam were condensed at each filling, the air and vapor, unless they were removed, would accumulate from these various sources until the vacuum would be entirely destroyed.

10. Air and vapor differ from water in that they are expansible, while water is not. To illustrate this, suppose that two closed vessels *A* and *B*, Fig. 2, are filled, *A* with water and *B* with air or vapor; now pump out, say, one-half

the water in A, as shown at C. The space a above the water is a vacuum; but if one-half the air or vapor in B is pumped out, the air or vapor will still fill the vessel, but at a lower pressure. The air or vapor will have become attenuated, or thin, by expansion. In obedience to the law that the different pressures of a gas are inversely to their volume, the pressure in D will be just one-half what it was in B before one-half the air or vapor had been withdrawn. If the original pressure in B was 15 pounds absolute per square inch, it will now be $7\frac{1}{2}$ pounds absolute per square inch in D, or since $7\frac{1}{2}$ pounds pressure correspond closely to 15 inches of mercury, we may say that a vacuum of 15 inches exists in the vessel. Should three-fourths of the air and vapor in B be pumped out, the pressure will be one-fourth the original pressure, that is, $3\frac{3}{4}$ pounds, and $15 - 3\frac{3}{4} = 11\frac{1}{4}$ pounds of the pressure have been removed. Since a vacuum gauge indicates not the pressure that exists in the vessel, but the pressure that has been *removed* from the vessel, counting from the pressure of the atmosphere, a vacuum gauge would now indicate $11\frac{1}{4} \times 2 = 22\frac{1}{2}$ inches of vacuum, nearly, because a pressure of 1 pound per square inch corresponds very nearly to 2 inches of mercury indicated by the vacuum gauge.

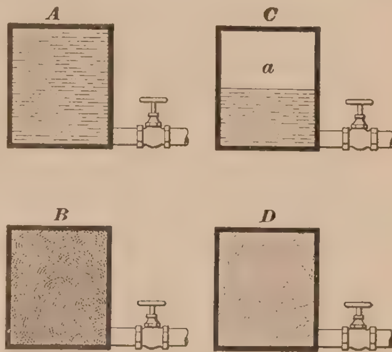


FIG. 2.

11. The object of the condenser is to remove a large part of the back pressure on the exhaust side of the piston of a steam engine when exhausting into the atmosphere, which back pressure, obviously, cannot be less than the pressure of the atmosphere. By making the engine exhaust into a condenser, the back pressure will be lowered to the

pressure existing in the condenser, and, consequently, with the pressure on the steam side of the piston remaining the same as before, the net pressure on the piston will be increased by the use of a condenser. As previously explained, air and vapor will collect in the condenser and if not removed will destroy the vacuum. To get rid of this air and vapor, the condenser is fitted with an air pump, or is provided with other means by which the air and vapor are removed from the condenser along with the condensed steam and condensing water. This operation restores and maintains a constant vacuum.

LOSSES IN CONDENSING ENGINES.

12. A condensing engine requires more watchful attention on the part of the running engineer than an engine exhausting into the atmosphere. Without this extra attention the condenser may be the source of a positive loss, so far as the consumption of steam is concerned. Leaky steam valves and leaky pistons are largely responsible for this loss. Leaks of this nature are not so easily detected in the condensing engine as they are in the non-condensing engine, with which the exhaust steam is usually in view as it leaves the exhaust pipe. A continuous stream of steam being emitted from the exhaust pipe indicates leaky steam valves or leaky piston or both; the only sure way to discover such leaks in a condensing engine is by means of the indicator card.

13. There may be another source of loss in a condensing engine, when independent air and circulating pumps are used, in the extravagant use of steam in the engine or engines that operate these pumps. The steam cylinders of the air and circulating pump should be fitted with indicator gear and cards taken, and the steam valves should be set with the same care that is bestowed upon the main engine. In large plants a considerable saving of steam may be effected by compounding the steam cylinders of independent

air and circulating pumps, or in the case of multiple-expansion engines, to lead the exhaust steam from the engines of independent air and circulating pumps into the receiver. By so doing, the energy remaining in the exhaust steam may be converted into work in the low-pressure cylinder of the main engine.

CONSTRUCTION OF CONDENSERS.

CLASSIFICATION.

14. Condensers may be divided into two general classes, which are *jet condensers* and *surface condensers*.

15. A **jet condenser** may be defined as a condenser in which the exhaust steam and the condensing water mingle and where the steam, consequently, is condensed by direct contact with the water.

16. In a **surface condenser** the exhaust steam and the condensing water do not mingle; the exhaust steam is condensed by coming into contact with metallic surfaces that are kept cool by cold water constantly flowing over them.

JET CONDENSERS.

TYPES.

17. There are three types of jet condensers, viz.: *The common jet condenser, the siphon condenser, and the induction condenser.*

THE COMMON JET CONDENSER.

18. The **common jet condenser** consists of an air-tight vessel of sufficient strength to sustain the pressure of the atmosphere from without. The exhaust steam flows into this vessel from the cylinder after its available energy has

been spent in driving the piston to the end of its stroke. The steam on entering the condenser is met by a jet or spray of water forced through small apertures in a pipe placed in the pathway of the incoming steam. The steam and spray are thus mechanically mixed and the steam, surrendering much of its heat to the sprayed water, is condensed into water, which falls to the bottom of the condenser along with the injection water. From the condenser it flows through the foot-valve into the air pump, by which it is pumped into the hotwell. So much of the water as may be required for feeding the boilers is taken from the hotwell by the feed-pump and forced into the boilers, the remainder of the water in the hotwell flows through the discharge pipe into the sewer or elsewhere.

19. The form of a common jet condenser is immaterial, but it is important that it be placed so that the water cannot be drawn into the cylinder from it in case of flooding.

20. The common jet condenser in its simplest form is shown in Fig. 3, in which *A* is a cylindrical or rectangular

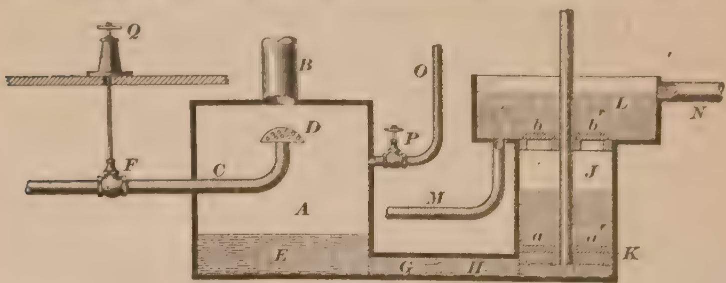


Fig. 3.

vessel forming the **body** of the condenser; *B* is the exhaust pipe from the engine through which the exhaust steam enters the condenser; *C* is the **injection pipe**; *D* is a perforated enlargement on the end of the injection pipe, called a **rose**, whose duty it is to scatter, or spray, the injection water that is forced into the condenser by the pressure of the atmosphere on the surface of the water supply outside.

The spray coming into direct contact with the incoming steam deprives the steam of a part of its heat and condenses it into water, which, together with the injection water, falls to the bottom of the condenser, as shown at *E*; *F* is the **injection valve,** which regulates the flow of the injection water into the condenser.

21. The handle or wheel *Q* of the injection valve of a jet condenser is always placed within reach of the engineer's station, as the valve must either be opened or closed at the same moment that the engine is, respectively, started or stopped; otherwise the condenser either will be flooded with water or else it will get hot.

Should the condenser become flooded there is danger of the water being drawn into the cylinder and cause serious damage by blowing out the cylinder head or breaking the piston; besides, the condenser being full of water when flooded, there is no room for the exhaust steam to enter and a vacuum cannot be formed until the air pump clears the condenser of its superfluous water, which will require several strokes of the bucket to accomplish. On the other hand, should the condenser be deprived of injection water the incoming steam cannot be condensed, and it will accumulate in the condenser until the pressure is equal to or greater than the pressure of the atmosphere outside; the condenser is then said to be hot. When this occurs, the injection water cannot be forced into the condenser by the pressure of the atmosphere alone. To provide for such a contingency, jet condensers of this type are usually fitted with a water pipe connected with one of the auxiliary pumps, by which cold water is forced into the condenser and a vacuum is produced thereby. This pipe is shown at *O*, Fig. 3, and is provided with a valve *P*, which should be kept closed when not in actual use, to exclude any air that might leak into the condenser through the auxiliary pump. If no such pipe is provided, it will be necessary, in case of the condenser becoming hot, to deluge the outside of it with cold water from a hose or buckets.

22. A **foot-valve** *G* is placed in the channel way *H* through which the water flows from the condenser into the air pump; *J* is the **air-pump barrel**; *K* is the **air-pump piston,** or **bucket,** as it is commonly called. The bucket is perforated with apertures that are fitted with flat valves *a*, *a'*, which open upwards; when the bucket descends the valves open, permitting the air, vapor, and water in the lower, or receiving, chamber of the air pump to pass through the openings in the bucket into the barrel of the pump. The bucket, in ascending, forces the air, vapor, and water through the **delivery valves** *b*, *b'* into the **hotwell** *L*. Whatever quantity of water may be required for boiler feeding is taken from the hotwell by the feed-pump through the pipe *M*; the remainder of the water in the hotwell runs off through the **discharge pipe** *N*.

If the injection water is so impure as to be unfit for boiler feedwater, all the water in the hotwell is allowed to run to waste and pure feedwater is supplied from another source; this is a great loss of heat, however, as the temperature of the feedwater must be raised to at least that of the water in the hotwell to make it of equal value as feedwater; to do this, extra heat is required from the fuel, unless the exhaust steam from the auxiliary engines is utilized in a feedwater heater for that purpose, in which case the assistance of the vacuum is lost to the auxiliary engines. This statement applies to all condensers in which the injection water comes in direct contact with and is mixed with the exhaust steam.

23. While the construction of a jet condenser shown in Fig. 3 is practically obsolete at present, it clearly shows all the essential parts, and its simplicity renders its operation easily understood. With this fundamental knowledge of its operation, it is comparatively easy to understand the action of later designs of jet condensers.

24. The jet condenser now generally used differs somewhat in form from the one just described. It occupies less

space and the air pump used in connection therewith is usually a double-acting, horizontal pump operated by an independent engine, but the principles involved are practically the same in both. As an example of this type of condenser a description of one of this class is given below.

In Fig. 4 is shown a section of a Worthington independent jet condenser. The cold water enters the condenser at *b*,

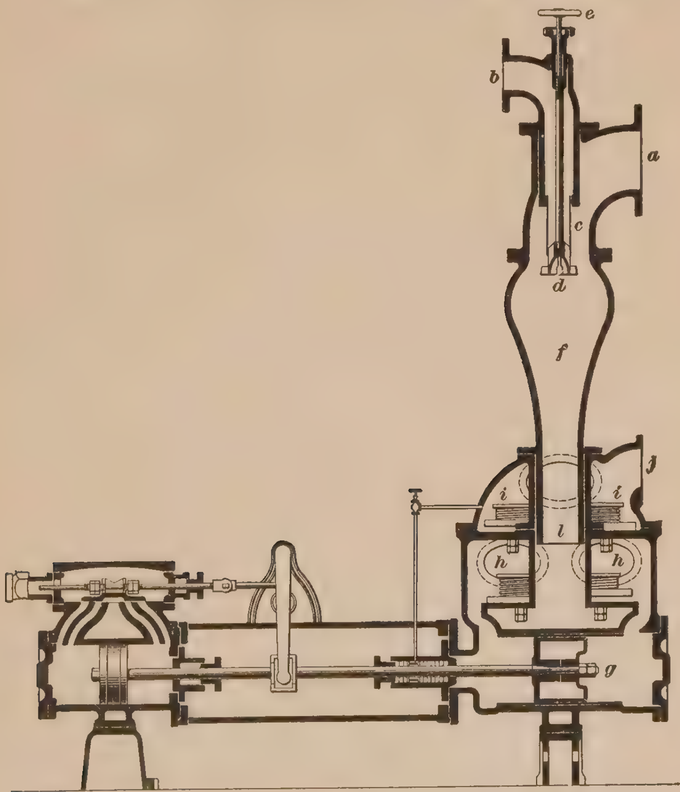


FIG. 4.

passes down the spray pipe *c*, and is broken into a fine spray by the cone *d*. The exhaust steam in the meantime comes in at *a*, and, mingling with the spray of cold water, is rapidly condensed. The velocity of the entering steam is imparted to the water, and the whole mixture of steam,

water, uncondensed vapor, and air is carried with a high velocity through the cone *f* into the air-pump cylinder *g*, whence it is forced by the pump through the discharge pipe *j*.

25. The above described condensing apparatus is operated as follows: The air pump having been started, a vacuum is formed in the condenser, the exhaust pipe, the engine cylinder, and injection pipe; this causes the injection water to enter through the injection pipe attached at *b* and to flow through the spray pipe *c* into the condenser cone *f*. The main engine being then started, the exhaust steam enters through the exhaust pipe attached at *a*, and, coming into contact with the cold water, is condensed. The velocity of the steam is communicated to the water and the whole passes through the cone *f* and through the receiving valves *h*, *h* into the pump *g* at a high velocity, carrying with it in a commingled condition all the air and uncondensable vapor that enters the condenser with the steam. The mingled air, vapor, and water are expelled by the pump through the discharge valves *i*, *i* and the delivery pipe at *j* before sufficient time or space has been allowed for separation to occur.

The spray pipe *c* has at its lower end a number of vertical slits through which the injection water passes and becomes spread out in thin sheets. The spray cone *d*, by means of its serrated surface, breaks the water passing over it into fine spray and thus insures a rapid and thorough admixture with the steam. This spray cone is adjustable by means of a stem passing through a stuffingbox at the top of the condenser and is operated by the handle *e*.

26. Exhaust steam from an engine enters a vacuum with a velocity of about 1,900 feet per second, and water, under atmospheric pressure, with a velocity of 47 feet per second.

In the common jet condenser, the injection water and the condensed steam fall to the bottom of the condensing chamber and come to rest there before entering the air pump;

thus the momentum acquired by water rushing into a vacuum is lost; whereas, in the class of jet condenser now under consideration, this force is utilized to assist the pump by accelerating the velocity of the falling water. This is accomplished by contracting the lower end of the condenser cone into a neck, or throat, as shown at *l*, Fig. 4. Moreover, the air and vapor being intimately mixed with the water, the load on the pump is more steady and regular than is the case in the common jet condenser.

The danger of the water working back into the engine cylinder, in case of flooding, is less with this form of jet condenser than with the common jet condenser.

27. The injection opening of a jet condenser must not be more than 20 feet above the surface of the water supply, and it is highly important that the injection pipe be entirely free from air leaks.

28. In Fig. 5 is shown a jet condenser in connection

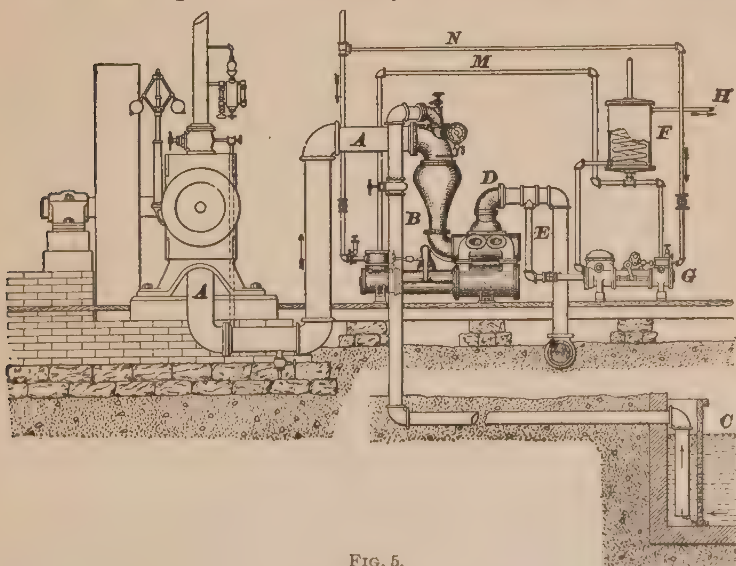


FIG. 5.

with the boiler and engine. The exhaust pipe *A* leads directly to the condenser. The injection pipe *B* draws

water from the reservoir *C*. After the steam is condensed, the mixture of exhaust steam and injection water is discharged through *D* into the sewer. A portion of this discharge, however, flows through *E* to the feed-pump *G*, which forces it through the coil in the heater *F* to the pipe *H* leading to th boiler. The exhaust from the two pumps is discharged into the feedwater heater through the pipe *M*. It will be noticed that water from the overflow pipe *D* enters the feed-pump under a slight head. This is because the water is heated by the exhaust steam, and hot water cannot be raised by a pump like cold water. A pipe *N* leads from the boiler and supplies steam for both pumps.

THE SIPHON CONDENSER.

29. The **siphon condenser** differs from the common jet condenser in that no air pump is required to remove the air, uncondensed vapor, and water, but a circulating pump or a head of water is needed to supply the injection water when the lift is more than 20 feet. The vacuum is generated and maintained by a column of water flowing downwards through a vertical pipe of not less than 34 feet in length, having its lower end immersed in the water of the hotwell.

30. It will be remembered that a column of mercury 30 inches in height or a column of water 34 feet in height will just balance the atmosphere at the sea level when the barometer stands at 30 inches, but if an additional amount of water or mercury be allowed to enter the upper end of the water pipe or mercury tube, the equilibrium between the column of water or mercury and the column of air outside will be disturbed, and an amount of water or mercury corresponding to that allowed to enter at the upper end of the pipe or tube will flow out at the lower end.

This is the principle of the siphon condenser. So long as the proper amount of water continues to flow into the upper end of the pipe and a corresponding amount flows out at the lower end, the air and vapor in the condenser will be carried

out by the descending water and a vacuum will be formed and maintained therein. If the area of the pipe is contracted into a neck, or throat, the velocity of the falling water will be accelerated and the action of the condenser will be improved thereby.

It is important that the stream of injection water entering the condenser should have a steady and continuous flow, and there must be no air leaks in the exhaust pipe or condenser. The siphon condenser is often, but wrongly, called the **injector condenser.**

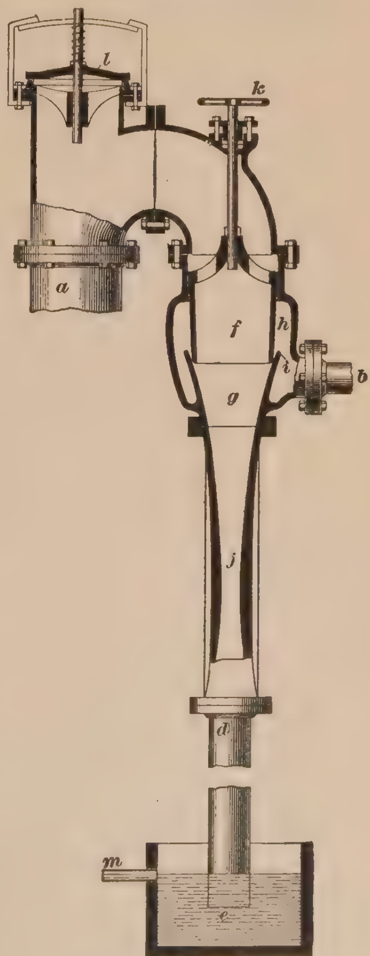


FIG. 6.

31. An illustration and a description of an example of this type of condenser, known as the **Baragwanath condenser,** is here given.

Fig. 6 represents a sectional view, in which *a* is the exhaust pipe; *b* is the injection pipe; *d* is the long discharge pipe, or **tail-pipe,** and *e* is the hotwell. The operation of this condenser is as follows: The steam enters through the exhaust pipe *a* and flows through the exhaust nozzle *f* into the condensing chamber *g*. Here it is met and condensed by the injection water that enters from the water-jacket *h* into the condenser in a thin conical sheet, flowing through the annular opening between the exhaust nozzle *f* and the

prolongation of the shell of the condenser forming the inverted cone *i*. A vacuum is now formed in the condensing chamber *g* by the condensation of the steam and by the air and uncondensed vapor being entrapped and carried out of the chamber by the cylindrical stream of water. The injection water and the water of condensation flow from the condensing chamber *g* down through the throat *j* with such velocity as to carry with them the air and vapor that passed over with the steam and the injection water.

The exhaust nozzle *f* is adjusted by means of the wheel and screw spindle *k*, and can be set so as to admit just the right quantity of injection water. An automatic atmospheric relief valve *l* is fitted for the purpose of discharging any excessive accumulation of air, steam, or vapor that may collect in the exhaust pipe into the atmosphere. A hotwell overflow or discharge pipe *m* is always fitted to the hotwell.

Although the condenser is placed at a height of 34 feet above the hotwell, the vacuum assists the circulating pump in a proportionate degree, so that with a vacuum of 24½ inches the actual height that the water is forced by the pump is but 6 feet.

32. It is sometimes the practice with this type of condenser to place a tank about 15 feet below the condenser from which the injection water is forced into the condenser by the pressure of the atmosphere upon the surface of the water in the tank, or *siphoned in*, as it is termed. The tank is supplied with water by an ordinary tank pump or from the street main. This arrangement has two advantages, viz.: (1) It insures a steady flow of water into the condensing chamber, which is an important consideration, and (2) a lower priced circulating pump may be used. In this case, however, there must be a cross-pipe connection between the injection pipe and the discharge pipe at the water-supply level, for the purpose of charging the siphon and generating a vacuum in the condenser. This is done by opening the valve in the cross-pipe connection and admitting water to

the discharge pipe; the water in falling down the discharge pipe draws the air from the condensing chamber and a partial vacuum is formed, which induces the injection water to enter. As soon as the siphoning action through the condenser is established, the valve in the cross-pipe connection is closed.

33. If the level of the injection-water supply is not more than 20 feet below the injection inlet to the condenser, the water will be siphoned over as soon as a vacuum is formed, and a circulating pump may then be dispensed with.

34. The flow of water, impelled by the incoming steam and by its own gravity, pours through the throat *j* continuously in one direction and with such high velocity that the possibility of the condenser becoming flooded and the water working back into the engine cylinder is very small.

The exhaust nozzle being adjustable admits of close regulation of the injection-water supply, which still maintains the conical form of the stream of water as it enters the condensing chamber and exposes the same area of condensing surface; even when nearly shut off there are no bare spots on the sides of the condenser, the conical sheet of water simply being made thinner.

THE INDUCTION CONDENSER.

35. The operation of the **induction condenser** is based on the same principle as that of the steam injector, used so largely for boiler feeding, and it may properly be called an **injector condenser,** although it is not given this name by the trade.

Fig. 7 represents a partial sectional view of a condensing apparatus of this type. It is known as the **Korting universal exhaust steam induction condenser** and is manufactured by L. Schutte & Co., Philadelphia, Pennsylvania.

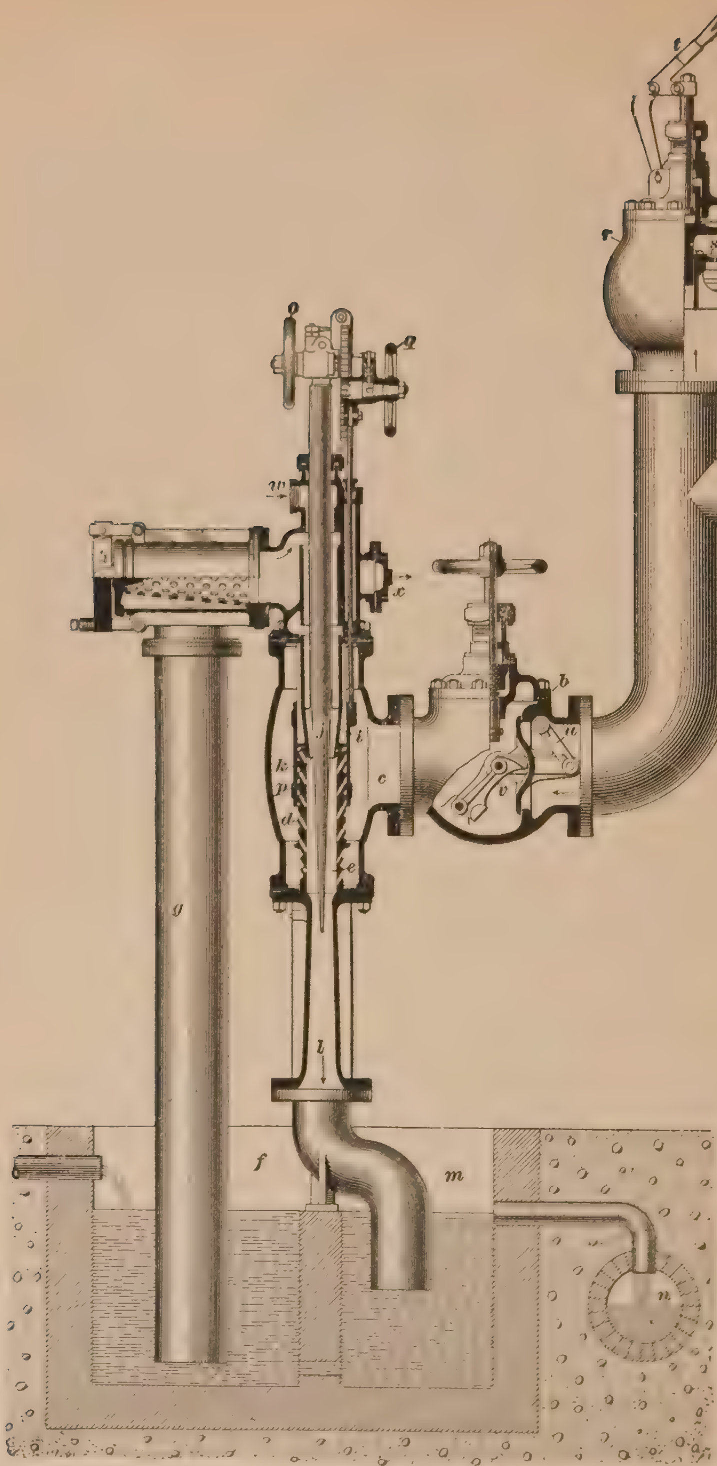


FIG. 7

Referring to the figure, the exhaust steam enters at *a*, and after passing through the balanced horizontal check-valve *b* enters the water chamber *c*; it then passes through the inclined openings in the tube *d* into the condensing chamber *e*, where it is met by the injection water and is condensed, forming a partial vacuum—the condenser having first been started by a supplementary jet of steam or stream of water—as will be explained hereafter. The vacuum in the chamber *e* induces the injection water to be siphoned into the condenser from the supply reservoir *f* through the injection pipe *g* and the strainer *h*, from whence it flows into the annular space *i* around the ram *j*, passing into the condensing chamber *e* through the annular opening *k*, where it meets the exhaust steam, which is then condensed. Here the injection water and the water of condensation intermingle and with the air and vapor are carried down the discharge pipe *l* into the hotwell *m*, the surplus water flowing into the sewer *n*.

36. To obtain the best results under the varying quantities of steam it may be called upon to handle, this condenser requires that it shall be adjustable. This is accomplished by the ram *j* being made tapering and capable of being raised and lowered at will, which operation varies the size and capacity of the annular opening *k* and controls the volume of water admitted to the condensing chamber *e*. The ram is adjusted by the hand wheel *o* acting through a rack and pinion. The area of opening required by the steam that enters the condenser is regulated by the sleeve *p*, which covers more or less of the openings in the tube *d*, as may be required. This sleeve is raised or lowered by the hand wheel *q*, which also acts through a rack and pinion. It will be observed that this apparatus is capable of very fine adjustment.

37. Like all condensers, this one requires a valve that opens into the atmosphere to relieve it of any accumulation of steam, air, or vapor that may collect in it. This is

provided in the **automatic free exhaust valve** *r*. This valve closes automatically when the condenser contains a vacuum and opens automatically when the vacuum is destroyed. It is fitted with a piston *s* to prevent the valve hammering. If it should become necessary or desirable to cut the condensing apparatus out of service and to run the engine non-condensing, the free exhaust valve may be locked open by turning the hand lever *t* to the left.

The operation of the check-valve *b* is as follows: The inclined suspension bar *u* has a tendency to open the valve, while the inclined supporting bar *v* has a tendency to close it. Thus the valve is balanced by its own weight, which is so distributed that there is always a slight excess of closing tendency. The object of this valve is to prevent the water in the condenser being siphoned back into the steam cylinder.

38. When the injection water is supplied to the condenser under pressure, as from an elevated tank or from the street main, instead of being siphoned up, the openings *w* and *x* are blanked.

When the injection water is siphoned into the condenser and water under pressure is used for starting, the starting water enters at the opening *w* and the opening *x* is blanked.

When the injection water is taken under high suction and steam is used for starting, the steam enters through the opening *w*, a check-valve is attached at the opening *x*, and an overflow pipe is connected with the check-valve to discharge free or into the discharge pipe *l*.

39. It will be noted that this condensing apparatus requires neither air pump, circulating pump, nor tail-pipe, as is the case with the siphon condenser; therefore, all the power that is gained by working the engine condensing is made available for useful work, instead of being offset by the power required to work the air and circulating pumps, as in other types of condensers.

SURFACE CONDENSERS.

PURPOSE.

40. When the injection water is so impure as to be unfit for feedwater, the condensed exhaust steam may be saved for feeding purposes by keeping it separate from the injection water. It is the purpose of the surface condenser to do this.

The surface condenser differs from the jet condenser in that the steam is condensed by coming into contact with the cold surfaces of numerous tubes through which cold water is being pumped, instead of coming into direct contact with the injection water. In this way the water of condensation and the circulating, or injection, water, are kept entirely apart, which permits the water of condensation to be used as boiler feedwater. Condensed steam is pure water, therefore it is in the best possible condition for feedwater.

PUMPS REQUIRED.

41. The surface condenser requires two pumps, viz., an air pump and a circulating pump. The duty of the circulating pump is to force the injection water through the tubes of the condenser; the air pump removes the air, vapor, and water of condensation from the condenser. There being two pumps connected with the surface condenser to do the work that is performed by one pump in the case of the jet condenser, they can be much smaller than when only one pump is used.

ADVANTAGES.

42. Though the surface condenser is more complicated, its first cost greater, and it requires more attention on the part of the attending engineer than does the jet condenser, the value of pure feedwater for the boilers more than compensates for these disadvantages. It is far better to keep

impurities out of the boilers entirely than to dose them with patent compounds, chemicals, kerosene oil, etc., or to allow the impurities to accumulate in the boilers in the form of scale, mud, etc. to the great detriment of the boilers, incurring loss of fuel and time and the expense of scaling and cleaning them.

CONSTRUCTION.

43. Construction of the Condenser.—Fig. 8 represents a sectional view of a surface condenser, without the

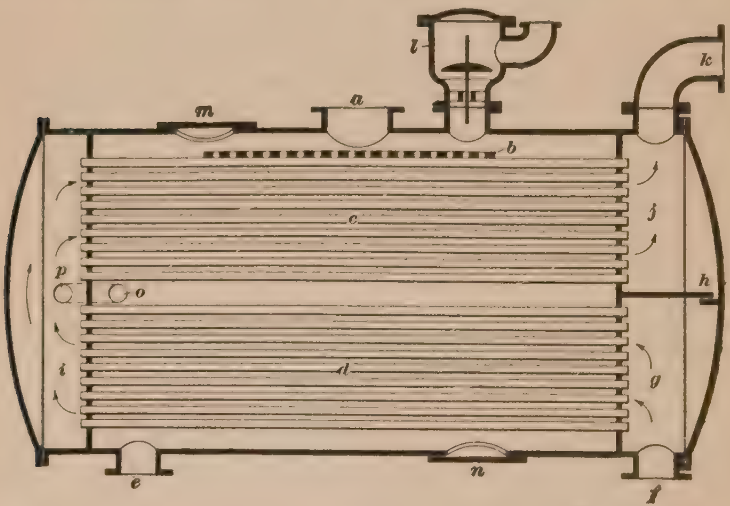


FIG. 8

pumps. The body of the condenser is an air-tight vessel of either cylindrical or rectangular form, it matters not which. The exhaust pipe from the engine is connected at *a*. The steam, on entering the condenser, strikes the perforated scattering plate *b*, which distributes the steam over the tubes more evenly than otherwise would be the case, and it also protects the upper tubes from the damaging effects of the exhaust steam. The steam then circulates around the tubes of the

upper nest of tubes *c*. The instant the steam comes into contact with the cold surfaces of the tubes it is condensed into water, which falls in the form of rain upon the lower tubes *d*, around which it circulates, and is further cooled as it falls to the bottom of the condensing chamber and flows out of the nozzle *e* to the air pump.

The injection water passes from the circulating pump through a pipe attached at *f* to the receiving chamber *g*; it is here impeded in its course by the division plate *h*, which deflects the water into the lower tubes *d*, through which it passes to the water chamber *i* at the opposite end of the condenser; from thence it returns through the upper tubes *c* into the delivery chamber *j* and then out of the discharge nozzle *k*. An automatic atmospheric relief valve, sometimes called the **snifting valve,** is shown at *l*. This valve opens outwardly and is for the purpose of relieving the condenser of any excess of steam, air, or vapor that may accumulate within it. It would also be useful in the event of the breaking down of the air pump or of its becoming inoperative from any cause, as the engine could then be run non-condensing by exhausting through this valve into the atmosphere.

44. Making Up for Loss of Water.—Part of the steam that is generated in the boilers is lost in various ways, such as blowing off with the safety valve, opening the gauge-cocks, and blowing the whistle, also through the cylinder and other drain cocks, by leakage, etc.; therefore, after all the exhaust steam from the engine is condensed into water there will not be enough of it to keep up the supply of feedwater; this deficiency is sometimes made up by drawing a corresponding amount of water from the circulating side of the condenser. For this purpose a U-shaped by-pass pipe is fitted around one of the tube-sheets and provided with a cock or valve; this allows a communication to be opened up between the steam and water sides of the condenser; the openings for this pipe are shown at *o* and *p*.

45. Though it is sometimes necessary to make up the deficiency of feedwater from the injection supply, it is objectionable to do so, for the reason that, sooner or later, the water in the boilers will become almost as impure as if injection water only had been used as feedwater, and the object of the surface condenser will be defeated. It will then be necessary to blow off some of the very impure water from the boilers and to replace it with a corresponding amount that is less impure. This will cause a serious loss of heat, as the water blown out has already been heated to the temperature corresponding to the pressure of steam carried, while the water pumped in to take its place must also be heated to the same temperature before it will be in the proper condition to be converted into steam. To prevent this loss, it is best to make up the loss of water with purified water or with distilled water, if circumstances permit this to be done.

46. Effects of Grease.—In course of time the tubes of a surface condenser become coated with grease on the steam side, carried over from the cylinder by the exhaust steam. Grease being a non-conductor of heat, the efficiency of the condenser is seriously impaired when the tubes are thickly coated with it, and to restore its usefulness the grease must be removed. This was comparatively an easy matter when animal or vegetable oils only were used for lubricating the pistons. The condenser was fitted with a special cock, called variously an **impermeator, alkali cock,** or **soda cock,** by which caustic soda or caustic potash were injected into the steam side of the condenser upon the grease-covered tubes. The alkali coming into contact with the grease converted it into soap, saponifying it, so to speak. The soap, being soluble, would be dissolved and washed out through the drain cock. This operation was assisted and hastened by having a small live-steam pipe enter the condenser near the bottom. The steam side of the condenser being filled with clean water up to and covering the top row of tubes and the alkali introduced, steam was let into the

condenser through the small pipe until the water boiled. This was called boiling out the condenser and it was a very efficient way to get rid of the grease. But since mineral oils have almost entirely superseded animal and vegetable oils for cylinder lubrication, the boiling-out process is no longer feasible, because alkalies have no effect upon mineral oils; therefore, the grease deposited upon the condenser tubes from mineral oils must be removed by hand. In order to be able to clean out a surface condenser by hand, the tubes must be removed. In drawing the tubes, the greater part of the grease is stripped off the tubes by the tube-sheet; this necessitates a man going inside the condenser to scrape the grease off the tube-sheet. Manholes *m* and *n*, Fig. 8, are provided for this purpose.

47. It is not an easy task to clean out a surface condenser having several thousand tubes; it is one of the most disagreeable duties a practical engineer has to perform. This grease is very sticky, resembling tar, and is very difficult to remove not only from the tubes and tube-sheets, but also from the skin and clothing of the operator. It can generally be softened by a liberal application of kerosene oil or gasoline, which facilitates its removal. If gasoline is used, it must be remembered that the vapor given off is highly inflammable, and hence no naked light should be used near the condenser.

48. In view of the loss of efficiency in the condenser and the loss of time and the labor of cleaning out the grease, it is most desirable that the grease should be kept out of the condenser. This may be accomplished by introducing an efficient grease extractor in the exhaust pipe. The grease extractor is an apparatus that is coming largely into use and which contributes greatly towards keeping condensers and boilers free from grease. It should be found in every well-equipped condensing steam plant as a matter of economy.

49. The Double-Tube Surface Condenser.—Another form of surface condenser that is now extensively used is

the **Wheeler double-tube surface condenser,** which is manufactured by the Wheeler Condenser and Engineering Company, of New York City.

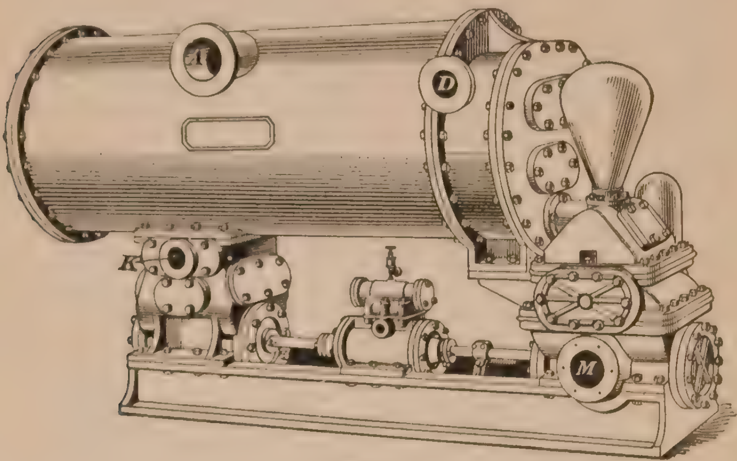


Fig. 9.

Fig. 9 is a perspective view and Fig. 10 a sectional view of the same. The injection water enters at *M*, Fig. 9, and

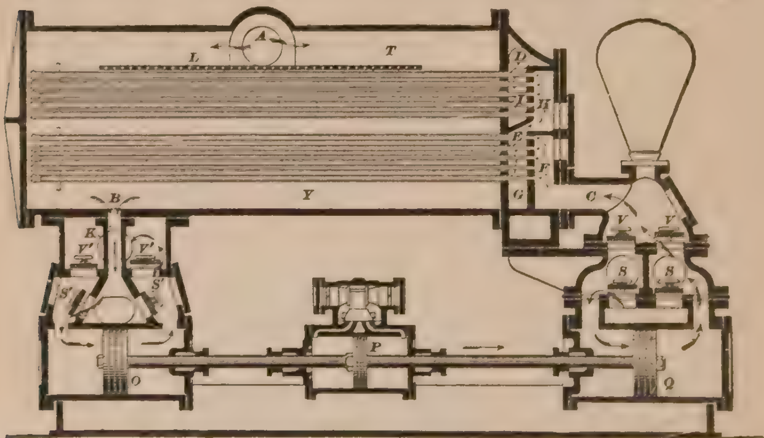


Fig. 10.

is forced by the circulating pump *Q*, Fig. 10, into the inlet *C* of the condenser. From *C* the water is forced into the

chamber F and flows, as indicated by the arrows, through the inner tubes of the lower nest of double tubing to the left. Having passed through their entire length, the water returns through the annular space between the outside of the inner tubes and the inside of the outer tubes into the chamber G. Fig. 11 shows more clearly the arrangement of this double tubing. From G, Fig. 10, the circulating water passes through E to H and from H to I through the upper

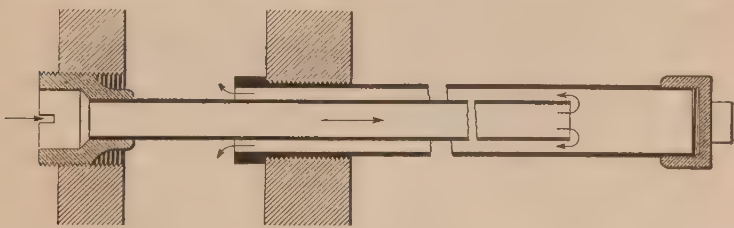


FIG. 11.

nest of double tubing, as has already been explained. From I it is discharged through the nozzle D, carrying with it all the heat it has taken from the exhaust steam while passing through the two nests of double tubing.

The nozzle A is connected with the exhaust pipe of the steam cylinder of the engine. The movement of the air-pump piston O draws the air, vapor, and water of condensation out of the condenser through the opening B and discharges them through the valves V', V' and nozzle K in the manner indicated by the arrows.

The valves S' and V' are opened and closed automatically by the pressure of the air beneath them and by the pressure of the air and springs above them. A partial vacuum is generated in the condenser Y by the air pump drawing the exhaust steam from the engine cylinder into the condenser, where it is condensed and a normal vacuum formed.

As the exhaust steam enters the condenser through the nozzle A, it first comes into contact with the perforated scattering plate L, which distributes the steam over the tubes and also protects the upper rows of tubes from the damaging effects of direct contact with the exhaust steam.

The steam then comes into contact with the cold tubes through which the cold circulating water is being pumped and is immediately condensed. As soon as this occurs, the water of condensation collects at the bottom of the condenser and flows through the opening *B* into the air-pump barrel, from which it is discharged by the piston of the air pump through the nozzle *K* into the hotwell or feed tank.

In this condenser the air and circulating pumps are operated by the independent steam cylinder *P*. The tubes of this condenser are secured to the tube-sheet at one end only and are free to expand and contract without danger of injury, rendering tube packing unnecessary.

DETAILS OF SURFACE CONDENSERS.

50. Condenser Tubes.—The latest practice regarding condenser tubes is to make them of a composition consisting of copper, 70 per cent.; zinc, 29 per cent.; tin, 1 per cent. They are $\frac{5}{8}$ inch outside diameter, No. 18 B. W. G. in thickness, and are spaced $\frac{15}{16}$ inch between centers. They should be carefully tinned within and without, and when they are 6 feet or over in length, they should be supported by one or more supporting plates. The tube-sheet should be 1 inch thick and made of the same composition as the tubes, with smoothly finished holes for the tubes and proper means provided for packing them.

51. Condenser-Tube Packing.—There is a number of methods of packing condenser tubes. The one generally used at the present time is shown in Fig. 12, in which *a* represents a part of the tube-sheet in section; *c* is a screw gland that screws down on the packing *d*. The packing may be either cotton tape, cotton lamp wicking, or a rubber ring. The screw glands are counterbored, forming the shoulder *e*, the purpose of which is to prevent the tubes crawling; the glands are slotted, as shown at *b*, to admit a tool for screwing up.

52. Supporting Condenser Tubes.—Condenser tubes are subjected to great changes in temperature and, in consequence, to extremes of expansion and contraction. If they were rigidly fixed in both tube-sheets, they would become warped, bent, and distorted, and would thereby soon be destroyed; hence the necessity of giving them end play.

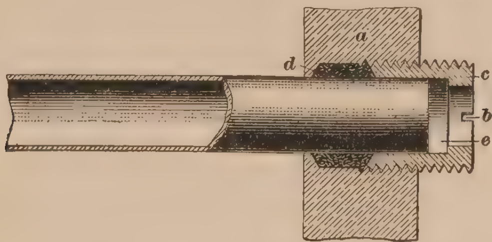


FIG. 12.

But the end play must be restricted, else the tubes will crawl by successive expansion and contraction until they become detached from either one or the other tube-sheet; this is obviated by the shoulder *c* in the gland. The space between the end of the tube and the shoulder is provided to give the tube room to expand or for the end play mentioned above.

LEAKAGE OF SURFACE CONDENSERS.

53. Water Leaks.—Owing to unequal expansion and contraction, the tubes of a surface condenser are subjected to very severe strains, and it is quite a common occurrence for them to split, causing leaks which if permitted to continue, will defeat the object of the surface condenser by admitting the injection water through the leaks to the steam side. The tube packing also will give out in time, which is another source of leakage. A serious leakage may make itself known by a noticeable increase in the feedwater, but these derangements must be looked for and the condenser tested for leaks occasionally or whenever the engineer in charge has reasons to believe it to be necessary. The test is made by removing

the condenser bonnets and filling the steam side of the condenser with water; both ends of every tube should then be carefully examined. If water flows from any of the tubes, it proves that those tubes are split, and they should be drawn out at once and new tubes inserted. If the time cannot be spared just then to put in new tubes, the leaky ones may be plugged up at both ends with dry white-pine plugs, as a temporary repair; but it is always best to make a permanent repair whenever possible. Defective tube packing should be looked for at this time and any packing that leaks should be renewed.

54. Air Leaks.—Air leaks in a condenser will be revealed by the vacuum falling, all other conditions being right. They can generally be traced by the whistling sound the air makes while being drawn into the condenser or they may be located by holding a lighted candle along the joints; the flame will be drawn inwardly at the leak. An air leak in a condenser can usually be stopped by setting up on the nuts of the bonnet or gland; if this does not stop the leak, plaster it over with red-lead putty containing a small proportion of litharge; this cement will harden in a short time and effectually stop the leak.

GALVANIC ACTION IN SURFACE CONDENSERS.

55. One of the impurities frequently found in water is sulphuric acid. This acid is derived from the decomposition of iron and copper pyrites (sulphurets of iron and copper), which are widely distributed throughout the earth, especially in mineral regions. The sulphuric acid—in connection with the copper in the condenser tubes and the iron of the condenser casing, feedpipes, and boilers—completes a galvanic battery, which sets up a galvanic current that attacks the iron or steel, causing rapid corrosion that is especially injurious to the boilers. To arrest the generation of this galvanic current, the condenser tubes are tinned

both inside and outside. The coating of tin prevents the acidulated water from coming into direct contact with the copper in the tubes.

56. As a further precaution against the deteriorating effects of galvanic action, plates of zinc are often suspended in the air-pump channel ways and in the hotwell or feed tank. The galvanic current attacks the zinc in preference to the iron or steel of the boilers and its destructive energy is expended in disintegrating the zinc. The zinc plates being gradually destroyed, it is evident that they must be renewed occasionally.

The metal straps by which the zinc plates are suspended should be filed bright where in contact with the zinc and in contact with the metal from which they are suspended. After the straps are bolted in place, the outside of the joints should be made water-tight by covering them with paint or cement to insure a good electrical contact.

The zinc plates are enclosed in wire baskets to prevent the larger particles of zinc that become detached from the plates during the progress of disintegration being carried into the boilers and lost by being blown out through the bottom blow.

CONDENSER ACCESSORIES.

COOLING TOWERS.

INTRODUCTION.

57. The great economy of using a condenser in connection with the steam engine has become so generally recognized by steam users, as it has been by engineers from the time of Watt, that it has been found desirable, as well as feasible, to install them in localities where the water supply is inadequate to operate a condensing apparatus in the ordinary manner. This demand has called into existence

various devices for the purpose of cooling the discharge water from a condenser and using it over again as injection water, which operation may be repeated indefinitely with the same water, needing only a small supply of extra water to replace that which is evaporated during the cooling process.

The principle on which all these cooling devices operate is that the evaporation of a part of the water undergoing the cooling process extracts the heat from the remaining part.

58. There are two distinct methods of cooling liquids; the first is by absorbing and carrying off the heat by a liquid previously cooled, as, for instance, occurs in the surface condenser, in which the heat of the exhaust steam is absorbed and carried off by the cold injection water; while the second method is effected principally by evaporation.

59. Evaporation is the conversion of a fluid into a vapor. When water is freely exposed to a current of air, the air in immediate contact with the water soaks up, as it were, more or less of the water and the air becomes charged with vapor. As evaporation proceeds only from the surface of liquids, the quantity of liquid evaporated depends on the extent of the surface exposed. Dry air absorbs moisture very rapidly at first, but more and more slowly as the process proceeds, until it ceases altogether and the air is then surcharged with moisture; it is then said to be saturated, which means that the air has soaked up all the moisture that it can hold. It is evident, then, that the more air that is brought into contact with the liquid to be cooled, the more rapidly will the evaporation proceed.

60. The capacity of air for absorbing moisture increases rapidly with the temperature. Dry air at 100° F. will take up a much greater quantity of moisture than air at 60°. One cubic foot of water vapor at a temperature of 60° F. weighs 5.745 grains, at 100° it weighs 19.766 grains, and at 110° it weighs 26.112 grains. If saturated air at 60° is

heated to 100°, it will no longer be saturated, and will then be capable of taking up 14.021 grains more moisture per cubic foot.

61. During the process of evaporation a great amount of heat is absorbed by the vapor, or, as it may be said, consumed in forming the vapor, and becomes latent. 966 B. T. U. are required to convert 1 pound of water into vapor at the atmospheric pressure. As this heat is derived or taken from the water that remains after a portion of it has been converted into vapor, the remaining water is cooled thereby.

CONSTRUCTION OF COOLING TOWERS.

62. Advantage has been taken of the cooling effect of evaporation to cool the discharge water of a condenser so that it may be used as injection water again. The application of the principle is usually through the medium of an apparatus called a **cooling tower,** which consists substantially, as its name implies, of a tower-like structure, usually about 30 feet high, to the top of which the warm water is pumped and there liberated and allowed to fall to the bottom of the tower against a current of ascending air which is impelled upwards by a fan blower. The water in descending is broken into spray or thin sheets by coming into contact with obstructions placed in its path for that purpose, thus presenting the greatest possible area of evaporating surface to be acted upon by the air.

There is a number of cooling towers in use, and while they differ somewhat in construction and material, they all are governed by the same principle.

63. The **Worthington cooling tower,** shown in Fig. 13, consists of a cylindrical steel shell *a* open at the top, supported upon a suitable foundation, and having fitted at one side a fan *b*, the function of which is to circulate a current of air through the tower and its filling. This filling consists of layers of cylindrical tubular tiling *c*, *c'*, *c''*, which rest upon

a grating *d* supported by a brick wall *e* extending around the circumference of the tower. The warm discharge water from the condenser enters the tower through the pipe *f*, passes up the central pipe *g*, and is delivered on the upper layer of tiling and over the whole cross-section of the tower by the distributing device *h*, which consists of four pipes, radiating from the central pipe *g*, which are caused to revolve about the central pipe by the reaction of jets of water issuing from perforations on one side of each pipe. The water thus delivered spreads over the outside and inside surfaces of the walls of the tiling and forms a continuous sheet, which is presented to the action of the air.

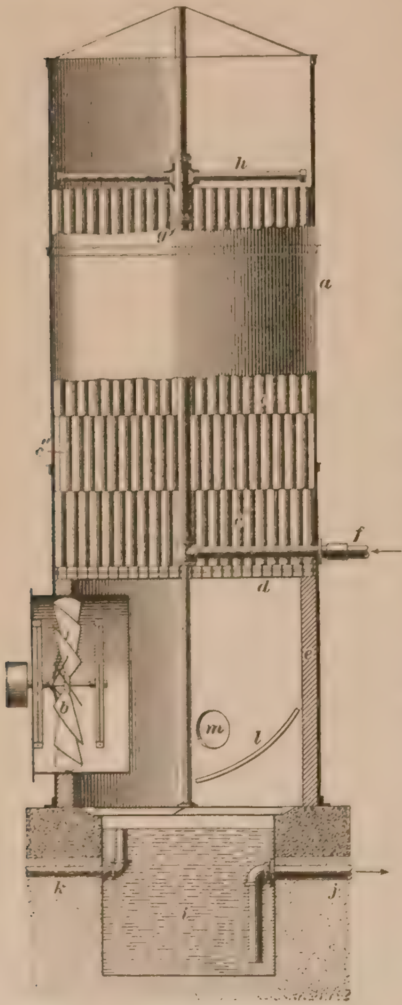


Fig. 13.

The tiling is placed on end in horizontal layers, one on the other, and is packed as closely as possible, the walls of each individual tile of each successive layer being disposed so as to come opposite the air space of the next lower layer, breaking joints, as it were; the object of this disposition is to break up both the currents of air and water so that the most extended contact will take place.

If there are ten layers of tiling in a tower, then there are nine places in addition to the original spreading at the top at which there is a complete redistribution of the water. It will be seen that each tile must rest on at least two, and possibly three, in the next lower layer. Assuming, however, that each tile in the upper row of tiles rests on only two others, a given quantity of water placed on any one tile in the top layer will be divided over at least two tiles in the second layer, three in the third, four in the fourth, and so on until it becomes spread over fifty-four tiles in the bottom layer on the grating.

The air is distributed in a similar manner, but in a reverse direction, to the flow of water, and there is a large free area for its passage upwards over the entire cross-section of the tower. The warm water falling through the tower is cooled by three processes: first, by radiation of heat from the sides of the tower; second, by contact with cool air; and third, by evaporation. The latter is by far the most effective and important, for the reason that the evaporation of a pound of water in this way carries off 966 B. T. U. The cooled water falls from the grating into the reservoir *i* at the bottom of the tower, and from there is forced into the condenser, either by atmospheric pressure or by a circulating pump through the injection pipe *j*, to perform condensation again.

The current of air is impelled through the tower by the circulating fan *b*, driven either by a small steam engine, an electric motor, or by belting from the main engine or line shaft, as the case may be.

A deflecting plate *l* is used to direct the current of air upwards through the tower. A manhole *m* affords access to the interior of the tower for examination, cleaning, and repairs, and *k* is the overflow pipe from the reservoir.

64. The **Barnard-Wheeler water-cooling tower** is represented in Fig. 14. The tower casing is usually constructed of steel plates. Within the tower are hung a number of mats made of special steel-wire cloth, galvanized

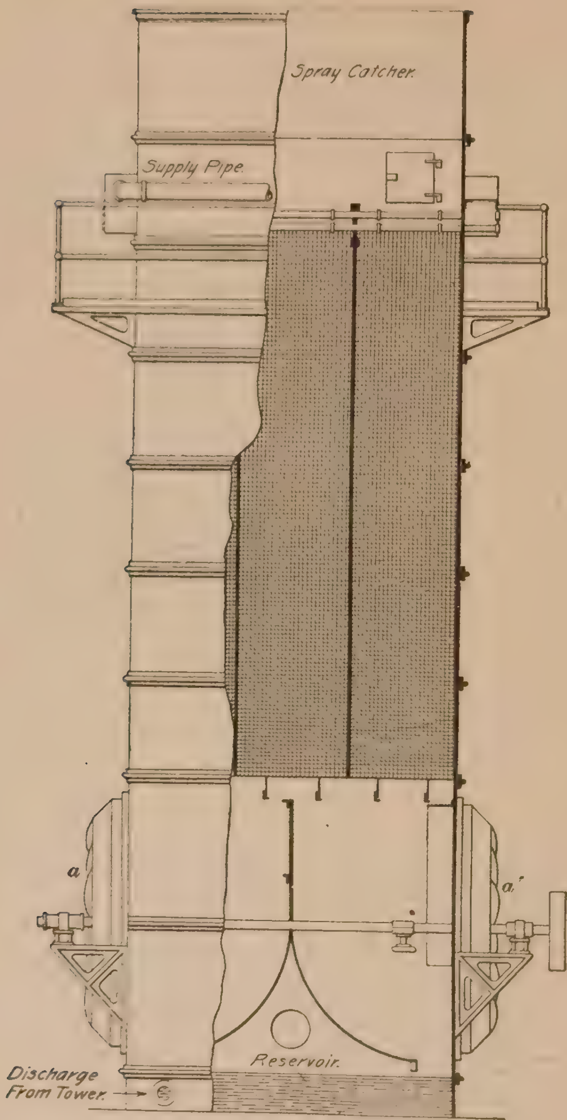

Spray Catcher.
Supply Pipe.
a
a'
Reservoir.
Discharge
From Tower.

FIG. 14.

after weaving. The pump discharge is led to the top of the tower and the warm water is there distributed by a suitable system of piping to the upper edges of the mats, over the surface of which it spreads in thin films, compelling a partial interruption of the flow and continuously bringing new portions of the water to the surface, thereby exposing it to the evaporating and refrigerating effects of the air-currents.

The mats are practically metallic sponges capable of holding a large quantity of water in suspension, which accumulates and drips off into the supply reservoir at the bottom of the tower. To assist the cooling action, the air in immediate contact with the water is set in rapid circulation by means of the fan blowers *a*, *a'*, Fig. 14, which force air into the lower part of the tower and upwards between the mats.

65. Barnard's fanless self-cooling tower is shown in Fig. 15. In this device the use of mechanical means for circulating air for cooling the water is dispensed with, thus avoiding the wear and tear and the expenditure of power that are always associated with moving parts. The hot circulating water discharged from the condenser is pumped up through the central stand pipe *a*, from which it is led to the trough *b* and distributing pipes *c*, *c*, causing a constant flow of thin films of water over the meshes of the wire mats *d*, *d'*, and finally draining into the tank or reservoir *f* forming the foundation of the tower, from whence the cooled water is returned through the injection pipe *e* for use again in the condenser.

The mats are placed radially and are entirely exposed to the atmosphere; they are so arranged as to permit the air to come into contact with the descending films of water by natural circulation, and the consequent evaporation is carried far enough to reduce the temperature of the injection water to a sufficiently low degree for condensing purposes.

66. The **Dean refrigerating tower** is rectangular in form and the cooling surfaces consist of metal tubes placed

horizontally, over which the water to be cooled slowly flows, dripping from tube to tube and exposing it in thin films to the refrigerating effect of an upward current of air kept in circulation by a fan blower.

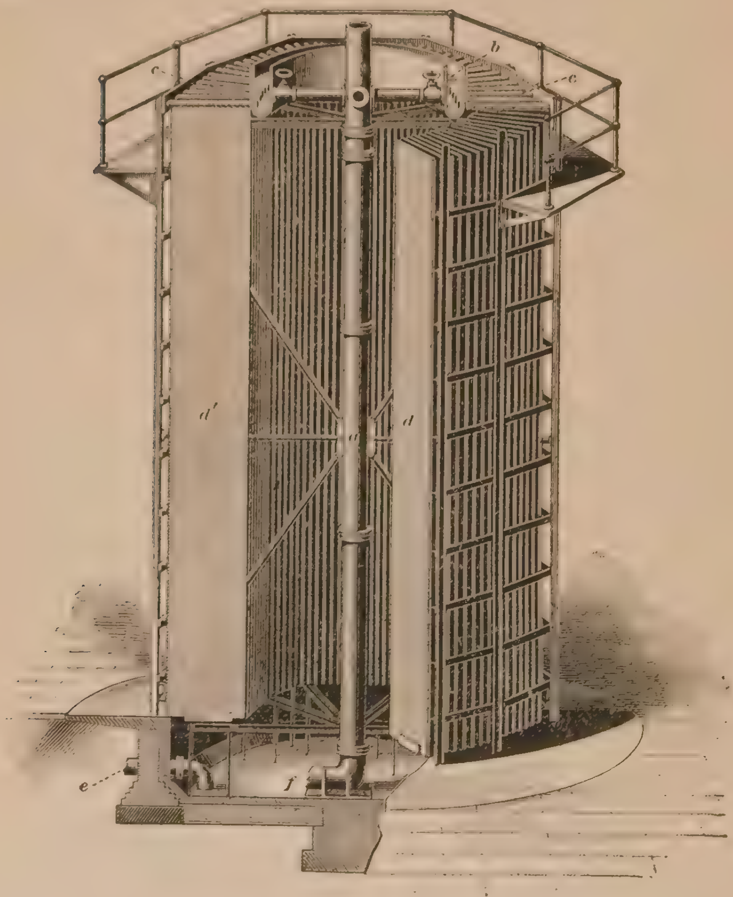


FIG. 15.

67. The **Stocker cooling tower** is a structure built of wood, steel, or bricks, according to circumstances. The cooling surfaces are built up of checkerwork or crosspieces of boards in horizontal layers set at right angles to one

another. At the intersections are placed upright partitions diagonally across the square openings between the boards. The water is pumped to the top of the tower and trickles down over these surfaces in thin films, which are broken up in falling at each intersection of the boards, and the water is thus brought into contact with the current of air that is forced upwards through the tower by fan blowers.

68. Cooling towers may be located on the roof of a building, if space for them is not available upon the ground, and they may be used in connection with any type of condenser. It is to be observed that there is a possibility of the water in a cooling tower freezing during very severe weather.

VARIOUS COOLING SYSTEMS.

69. There are other methods of cooling discharge water from condensers for the purpose of using it over again as injection water besides by cooling towers.

70. The **Schutte system** of cooling discharge water consists of a series of centrifugal spray nozzles that project the warm water from the condenser into the open air by a whirling action in the form of spray, under a pressure of 15 pounds or more to the square inch. A tank, pond, or other catch basin is provided to retain the cooled water.

71. The **Linde system** of cooling discharge water consists of a number of sheet-iron cylinders immersed about one-third of their diameters in the water of the condenser. These cylinders are revolved slowly, thereby carrying up thin films of water adhering to their surfaces, which, on coming into contact with a current of air, are evaporated, producing the cooling effect. The condenser is merely a tank containing submerged pipes connected with the engine exhaust. The water in the tank is kept in constant motion by an agitator.

LOSS OF WATER BY EVAPORATION.

72. When injection water is recooled, the loss of water by evaporation is proportional to the amount of heat carried off and depends on the difference between the temperature of the water before and after cooling. For example, if 200,000 pounds of water is cooled down from 120° to 60° in a given time, 60 B. T. U. are absorbed from every pound of water that is cooled down; therefore, $200,000 \times 60 = 12,000,000$ B. T. U. are carried off with the vapor. Now, as it requires about 1,000 B. T. U. to vaporize 1 pound of water, $\frac{12000000}{1000} = 12,000$ pounds of the water, or 6 per cent., have been converted into vapor and lost, which amount must be restored to the volume of available injection water from some original source of supply.

CONDENSER FITTINGS.

73. Certain fittings are required on a condenser to make it complete. A vacuum gauge, as a matter of course, is a necessity on all condensers, and it should be placed in a conspicuous part of the engine room, where the engineer may consult it at any time.

74. A surface condenser should be provided with standard thermometers, one each for the injection water, the feedwater, and the discharge water. The thermometers should be permanently connected to the condenser by means of small pipes, through which streams of the various waters mentioned should be kept circulating around the bulbs of their respective thermometers so that the temperature of the waters may be noted at a glance. Inasmuch as the best results can be obtained only by having the temperature of the feedwater as high as possible and the temperature of the discharge water as low as possible, and as these temperatures can only be accurately determined by means of standard thermometers, their utility is obvious. The

careful engineer that keeps the condenser temperatures at their best points will save many a ton of coal, and to enable him to do this the thermometers ought to be supplied, as a means of economy. The condenser temperatures are regulated by the injection valve or by the speed of the circulating pump, if independent. By these means more or less circulating water is passed through the condenser tubes, as required.

75. It may happen in winter that the injection water is very cold and that not enough of it is required for condensation to fill the barrel of the circulating pump, which may cause slamming of the valves. If the circulating pump is independent of the main engine, the difficulty may be remedied by slowing down the pump; but if the pump is attached to and operated by the main engine, this cannot be done. To meet this contingency, a reverse air valve is fitted to the barrel of the circulating pump, which will allow a certain amount of air to enter the pump at each stroke and relieve the slamming. The amount of lift of the air valve is controlled by a screw stem and hand wheel; the valve may be permanently closed by the same means.

76. Another method of accomplishing the same purpose is to provide a communication, by a pipe or channel way, between the discharge and the receiving chambers of the circulating pump, fitted with a valve, which, on being opened, will permit a part of the circulating water to return to the receiving chamber of the pump instead of passing through the condenser tubes. This valve is called the **regurgitating valve,** because it permits the water to regurgitate back towards its source. By regurgitating is meant the flowing back and forth of a liquid.

77. Surface condensers are usually provided with an **alkali cock,** by which caustic soda or caustic potash may be injected into the condenser for the purpose of dissolving the grease that collects upon the tubes from the exhaust

steam. The alkali cock is simply an ordinary plug cock screwed into the top of the condenser over and near the center of the upper nest of tubes, and having a cup or funnel-shaped top into which the alkaline solution may be poured. As an accompaniment to the alkali cock, a small live-steam pipe is run into the lower part of the condenser for the purpose of boiling the alkaline water as an aid to dissolving the grease.

78. A by-pass pipe containing a cock or valve is fitted around one of the tube-sheets of a surface condenser, whereby any deficiency of feedwater from the condensed steam may be supplied from the circulating water, provided there is no other source of supply.

79. The discharge water and feedwater being intermingled in a jet condenser, only two standard thermometers are required—one for the injection water and the other for the feedwater.

80. All condensers should be provided with a sufficient number of drain cocks so located as to completely drain all the water out of them. They should also have manholes and handholes wherever practicable and wherever necessary to afford access to the interior for examination, cleaning, and repairs.

81. The hotwell, or feed tank—which is practically the same thing and constitutes a part of the condensing apparatus—should be provided with a glass water gauge so that the height of water within may be observed at all times. It is by observing the fluctuations of the water level in this gauge that the engineer is informed as to the sufficiency or insufficiency of the feedwater supply.

82. A **vacuum breaker** is a necessary adjunct of a jet condenser. This is a device that will automatically break the vacuum in the condenser in case of a sudden breakdown of the air pump, and will thus prevent the filling up of the

condenser and the subsequent filling of the cylinder with water, which is coupled with the danger of wrecking the cylinder. A vacuum breaker is usually constructed in the form of a float-operated valve attached to the condenser and opening towards the atmosphere; it is so placed that when the water in the condenser exceeds a certain height, it will lift the float, open the valve, and admit air, thus destroying the vacuum and preventing a further inrush of injection water.

QUANTITY OF WATER REQUIRED FOR CONDENSATION.

83. The number of pounds of condensing water required to condense a pound of steam depends on the initial and final temperature of the steam and the initial and final temperature of the condensing water.

84. A pound of steam after being condensed to water having the temperature at which it leaves the condenser contains a number of B. T. U. above 32° F. that is given by subtracting 32 from its temperature. But a pound of steam before condensation contains a number of heat units above 32° corresponding to its pressure. Consequently, the number of B. T. U. that must be abstracted is given by subtracting from the total heat of the steam the difference between the final temperature of the steam after condensing and 32.

Each pound of condensing water in passing from its initial to its final temperature absorbs a number of B. T. U. equal to the difference in the initial and final temperatures. Therefore, the number of pounds of water required will be the number of B. T. U. to be abstracted from a pound of steam divided by the number of B. T. U. absorbed by a pound of condensing water. Hence the following rule:

Rule 1.—*To find the weight of condensing water per pound of steam, from the total heat of a pound of exhaust steam subtract the difference between the final temperature of the*

condensed steam and 32; divide the remainder by the difference between the temperatures of the entering and departing condensing water.

Or,
$$W = \frac{H - (t_3 - 32)}{t_1 - t_2},$$

where W = number of pounds of water required to condense a pound of steam;

H = total heat of vaporization of 1 pound of steam at the pressure of the exhaust. This may be obtained from column 5 of the Steam Table;

t_1 = the temperature of departing condensing water;

t_2 = the temperature of entering condensing water;

t_3 = the temperature of the condensed steam upon leaving the condenser.

The pressure of the exhaust is to be taken as the pressure at the point of release in the cylinder. In a jet condenser the final temperatures of the cooling water and the condensed steam are the same; that is, $t_3 = t_1$.

EXAMPLE 1.—The steam exhausts into a surface condenser at a pressure of 4 pounds absolute. The temperature of the condensing water on entering is 60° and on leaving is to be 100°. The temperature of the condensed steam on entering the air pump is 140°. How many pounds of condensing water are required per pound of steam?

SOLUTION.—From the Steam Table, the total heat of 1 pound of steam at a pressure of 4 pounds absolute is 1,128.641 B. T. U. Then, applying rule **1**, we have

$$W = \frac{1,128.641 - (140 - 32)}{100 - 60} = 25.52 \text{ lb.} \quad \text{Ans.}$$

EXAMPLE 2.—Steam exhausts into a jet condenser at a pressure of 2 pounds absolute. The temperature of the condensing water is 60° and the temperature of the mixture as it enters the pump is 135°. How much condensing water is used per pound of steam?

SOLUTION.—Applying rule **1**, we have

$$W = \frac{1,120.462 - (135 - 32)}{135 - 60} = 13.566 \text{ lb.} \quad \text{Ans.}$$

CAUSES OF AN IMPERFECT VACUUM.

85. An imperfect, i. e., a low, vacuum is due to one or more of three causes, which are the amount of condensing water supplied may be insufficient; the air pump may be out of order; or there may be air leaks.

86. The probable cause may be ascertained easily if a *log* of the performance of the engine is kept. In this log the temperature of the hotwell, the temperature of the discharged condensing water, the initial temperature of the entering condensing water are entered, say, every hour. Should the vacuum in a surface condenser be found imperfect, the temperature of the hotwell and the discharged condensing water should be ascertained. If the temperature of both is found to be more than has been noted down in the log book for a more perfect vacuum, it would tend to show that not enough condensing water is supplied. If an increased supply of condensing water fails to improve the vacuum, although it lowers the temperature of the hotwell and of the discharged condensing water, the indications are that an air leak exists somewhere about the condenser or engine. If no air leak is found, the air pump must be examined. Usually one or more of the air-pump valves will be found either broken or in poor condition. If the temperature of the hotwell and of the discharged condensing water are the same as in the case of a more perfect vacuum, the indications are that there is either an air leak or that the air pump is partially disabled.

87. In a jet condenser the temperature of the hotwell and that of the condensing water is always the same, since the exhaust steam and condensing water mingle. Hence, the hotwell temperature remaining the same, as in the case of a more perfect vacuum, either an air leak or a partially disabled air pump is indicated. If the hotwell temperature is greater, it indicates an insufficient quantity of injection water.

88. The maximum vacuum attainable, theoretically, can never be more, in inches of mercury, than the height of the column of the barometer at the same time and place. Owing to mechanical imperfections of the condensing apparatus, about the best vacuum that may be attained in practice will be within 2 inches of the theoretical vacuum.

COMPOUND ENGINES.

ADVANTAGES OF COMPOUNDING.

HISTORICAL.

1. In 1781 Jonathan Hornblower constructed and patented an engine, similar to the present "compound," that had two cylinders of different sizes. Steam was first admitted into the smaller cylinder and then passed over into the larger, doing work on a piston in each. In Hornblower's engine the two cylinders were placed side by side, and both pistons acted on the same end of a beam overhead. The use of this early compound engine was abandoned on account of the suit brought by a Birmingham firm for infringing their patent, which applied to the use of a separate condenser and air pump. At the beginning of the nineteenth century the compound engine was revived by Woolf, with whose name it is often associated, and who expanded the steam to six and even to nine times its original volume in two separate cylinders.

ECONOMY IN USE OF STEAM.

2. The compound engine has a mechanical advantage over the single-cylinder engine with an equal ratio of expansion and of equal power, but it is doubtful whether this

advantage was clear to either Hornblower or Woolf when they designed their compound engines. This advantage of the compound engine over the simple expansive engine is that the pressure on the bearings is more even throughout the stroke and not so high at the beginning of the stroke. However, a far more important merit of the compound engine lies in a fact that neither of them discovered for many years, and which is that by dividing the whole range of expansion into two or more parts and performing the work in separate cylinders, an economy in the use of steam was effected.

3. If an attempt is made to obtain a very high ratio of expansion in a single-cylinder engine, we are confronted with the fact that the gain due to the high ratio of expansion is lost entirely or even overbalanced by the loss due to the initial condensation of the entering high-pressure steam, this loss being occasioned by the cooling of the cylinder walls during exhaust.

4. Let the temperature of the entering steam be 753° absolute and the temperature of the exhaust steam 613° absolute. Then, the fall in temperature during expansion will be $753° - 613° = 140°$. Now, the cylinder walls, especially in engines having a very slow piston speed and having the cylinder poorly covered, will cool off to nearly the temperature of the exhaust steam, and hence the incoming steam must give up part of its heat in order to heat the cylinder walls again. As a matter of course, the condensed steam represents nearly a total loss of heat. We say *nearly* because, owing to the fact that the cylinder walls do not cool *entirely* down to the temperature of the exhaust steam and also on account of the reduction of pressure, some of the steam condensed on entering is evaporated again near the end of the stroke.

5. Suppose, now, that the temperature of the incoming steam is 842° absolute and that the temperature of the

exhaust steam is again 613°; that is, that we have a higher ratio of expansion. We now have a temperature range of 842° − 613° = 229°. A little thought will show that with this enlarged temperature range, a good deal more initial condensation will occur in order to raise the temperature of the cylinder walls to 842° again. From this the conclusion may be drawn that increasing the range of temperature increases the loss due to cylinder condensation.

6. Having seen that a high ratio of expansion, i. e., a high temperature range in a single cylinder, means a great initial condensation and subsequent loss in economy, let us see what the effect will be if the expansion of steam takes place in successive stages in several cylinders through the same temperature range. Then, the total temperature range being the same, it is evident that the temperature range in each cylinder will only be a fraction of it, and this has been found to reduce the condensation loss considerably below that occurring in one cylinder having the same total temperature range. This fact has been established beyond doubt by numerous experiments and has led successively to the adoption of compound engines, then triple-expansion engines, and lately quadruple-expansion engines.

7. We will now consider some facts showing the reason why a division of the temperature range between several cylinders tends to reduce cylinder condensation. It takes an appreciable period of time to change the temperature of the cylinder walls from the temperature of the exhaust steam to that of the entering steam. In most cases, if not in all, the time during which the entering hot steam is in contact with the cylinder walls is so short that the incoming steam cannot heat them to its own temperature. Likewise, the time during which the cylinder walls are in contact with the exhaust steam is usually so short that they do not cool entirely down to its final temperature. Consequently, the fluctuation of the temperature of the cylinder walls is less than the temperature range of the incoming and exhaust

steam. It is a well-known fact that the *rate* at which heat passes from a hot body (the entering steam in this case) to a cold body (the cylinder walls in the case under consideration) depends on the difference in temperature between the two bodies. When the temperature difference is great, the rate of heat transmission is much larger, proportionally, than it would be if the temperature difference were small. In other words, the rate at which the transfer of heat takes place increases much faster than the temperature difference. It follows from this principle, when the difference in temperature between that of the incoming steam and that of the cylinder walls, previously cooled by the exhaust, is small, that less heat, in proportion to the temperature difference, will be given up to reheat the cylinder walls than would be the case for a larger temperature difference. Similarly, if the temperature of the exhaust steam is but little below that of the cylinder walls, less heat in proportion to the temperature difference will be lost by the cylinder walls. Now, by expanding steam successively in several cylinders, we not only decrease the temperature range in each cylinder, but also greatly decrease the fluctuation in temperature of the cylinder walls of each. Then, as the rate of heat transmission is proportionally much smaller for a small fluctuation of temperature, it follows that the sum of the condensation losses in the several cylinders will be smaller than the condensation loss occurring in a single cylinder having the same temperature range between the initial and final temperature.

8. Another probable cause of reduction of cylinder condensation by compounding is that the steam condensed in the high-pressure cylinder and reevaporated during the exhaust period enters the low-pressure cylinder as *steam;* hence, the heat given up in the high-pressure cylinder by the entering steam in heating the cylinder walls is not a total loss, as would be the case if, instead of exhausting into a second cylinder, exhaust took place into the atmosphere or into a condenser.

MECHANICAL ADVANTAGES OF COMPOUNDING.

9. So far we have considered only the saving effected in the cylinders of the engine, as it is there that the greatest difference is shown; but there are other advantages incidental to the use of the triple-expansion and compound engines that deserve attention and recognition. The pressures, and consequently the strains, in a triple-expansion or a compound engine are more evenly distributed throughout the stroke than they are in a simple engine, and the turning moment on the shaft is also more nearly equal; therefore, lighter working parts can be used with the same margin of safety and lighter flywheels to give the same regularity of speed. These lighter weights and more even strains mean less friction in the engine, and this in turn means a saving of coal.

10. In order to show how this is possible, we will consider two engines of equal horsepower, the one a simple

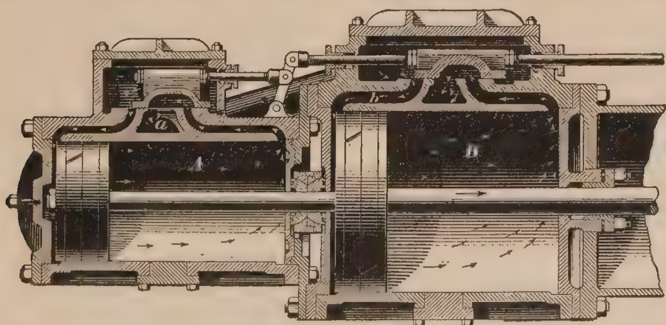


FIG. 1.

expansive engine and the other a so-called **tandem compound engine** that has two cylinders in line with each other and a piston rod common to both cylinders. Obviously, both pistons are carried on the same rod, as shown in Fig. 1. The cylinder *A* is called the **high-pressure cylinder** and the cylinder *B* is known as the **low-pressure cylinder.** The exhaust from the high-pressure cylinder passes through the exhaust passage *a* into the steam chest of the low-pressure cylinder.

The simple expansive engine has the following dimensions and data: Area of piston, 420 square inches; stroke, 30 inches; revolutions per minute, 90; steam pressure, 105 pounds gauge. The mean effective pressure of this engine, as found from the indicator card *A*, Fig. 2, is 48.66 pounds per square inch.

The tandem compound engine has the following dimensions and data: Area of high-pressure piston, 200 square inches; of low-pressure piston, 600 square inches; stroke, 30 inches; revolutions per minute, 90; steam pressure, 105 pounds gauge. The mean effective pressure, as found from the indicator card shown at *B*, Fig. 2, is 51.1 pounds in the high-pressure cylinder and 17.03 pounds in the low-pressure cylinder, as found from the indicator card shown at *C*, Fig. 2. The two engines will practically be of the same horsepower, since the mean impelling force on the piston of the simple engine is $420 \times 48.66 = 20{,}437.2$ pounds and the mean impelling force on the two pistons of the tandem compound engine is $200 \times 51.1 + 600 \times 17.03 = 20{,}438$ pounds, and their respective strokes and number of revolutions are equal.

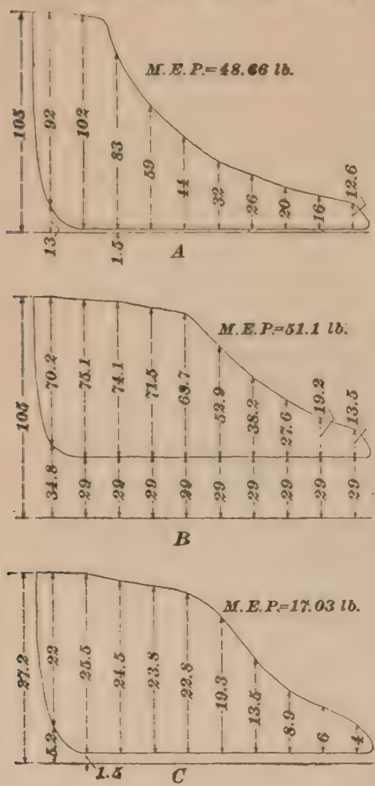


FIG. 2.

11. For convenience, we shall consider the load on the respective piston rods when the engines have made $\frac{1}{20}$ the stroke. Then, from the diagram *A*, Fig. 2, we find the net

pressure per square inch on the piston to be 92 pounds, the back pressure at this point being 13 pounds. It is to be understood that the ordinates, by means of which the mean effective pressures are found, are erected at points on the diagram corresponding to $\frac{1}{20}$, $\frac{3}{20}$, $\frac{5}{20}$, etc. of the stroke. Since the area of the piston is 420 square inches, the load on the piston rod at this point of the stroke is $420 \times 92 = 38,640$ pounds. In the tandem compound engine, by diagram *B*, Fig. 2, the pressure on the high-pressure piston is 70.2 pounds net; the net pressure on the low-pressure piston, by diagram *C*, Fig. 2, is 22 pounds. The reason these given pressures are the net pressures follows from what has been stated regarding the action of steam in the cylinders of a compound engine. Then, the load on the high-pressure piston is $200 \times 70.2 = 14,040$ pounds and the load on the low-pressure piston is $600 \times 22 = 13,200$ pounds, making a total load on the low-pressure piston rod of $14,040 + 13,200 = 27,240$ pounds. In the same manner, as shown above, we may calculate the loads on the pistons at $\frac{3}{20}$, $\frac{5}{20}$, $\frac{7}{20}$, etc. of the stroke and then arrange them in the form of a table, as given below:

Stroke.	Load on Piston Rod.	
	Simple.	Compound.
$\frac{1}{20}$	38,640	27,240
$\frac{3}{20}$	42,840	30,320
$\frac{5}{20}$	34,860	29,520
$\frac{7}{20}$	24,780	28,580
$\frac{9}{20}$	18,480	27,420
$\frac{11}{20}$	13,440	22,160
$\frac{13}{20}$	10,920	15,740
$\frac{15}{20}$	8,400	10,860
$\frac{17}{20}$	6,720	7,440
$\frac{19}{20}$	5,292	5,100
	204,372	204,380

From this table we see that while in the compound engine under discussion the maximum load on the piston rod is only 30,320 pounds, it is 42,840 pounds in the simple engine, thus showing that if the simple engine were converted into a compound of equal horsepower, retaining all parts except the cylinder and piston rod, the maximum load on the various parts, and hence the maximum stresses, would be less. An inspection of the table will also show that there is less variation in the loads in case of the tandem compound than in case of the simple engine; hence, the tandem compound will run steadier. The above shows that if conditions allow it, an engine giving trouble through weakness of its component parts and insufficient bearing surface may be remodelled and will probably give good satisfaction when changed to a tandem compound of equal power.

12. Having shown that the loads are smaller with a tandem compound engine, it will be seen upon reflection that an engine rigid in all its parts may have its horsepower increased by making it a tandem compound without exceeding the maximum loads the various parts sustained when it was a simple engine. Thus, if we changed the simple engine under discussion to a tandem compound giving indicator cards like *B* and *C*, Fig. 2, and having an area of 250 square inches in the high-pressure cylinder and 750 square inches in the low-pressure cylinder, the load on the low-pressure piston rod would be $250 \times 75.1 + 750 \times 25.5 = 37,900$ pounds at $\frac{3}{20}$ the stroke, or considerably less than if the engine were a simple expansive one. We will now compute the horsepower of both engines. That of the simple engine is $\frac{420 \times 48.66 \times 2.5 \times 2 \times 90}{33,000} = 278.7$ horsepower, nearly. The horsepower of the compound engine having cylinders of 250 and 750 square inches area is

$$\frac{250 \times 51.1 \times 2.5 \times 2 \times 90}{33,000} + \frac{750 \times 17.03 \times 2.5 \times 2 \times 90}{33,000}$$

$= 348.4$ H. P., nearly. We thus see that the horsepower

of the engine is increased by $348.4 - 278.7 = 69.7$ horsepower and that the maximum loads and stresses are still less than in case of the simple expansive engine here considered.

13. At one time or other, nearly every manufacturing concern finds that it needs more power. The common way of accomplishing this is to put in a larger cylinder, which, unless a very generous allowance of bearing surface has been made in the engine as originally constructed, leads, in a great many cases, to hot bearings and various other troubles. It would certainly seem to be the better plan to convert the engine into a tandem compound if conditions allow it. This is not always feasible, however; if the boiler pressure is less than 80 pounds gauge, it is hardly ever advisable to change to a compound. No general rule can be given as to when it is advisable to make the change; each case must be treated on its own merits.

14. So far only the tandem compound and the simple expansive engine have been compared, and it has been shown that the tandem compound engine possesses mechanical advantages over the simple engine of equal horsepower. Many compound engines are built as **cross-compounds,** one of which is shown in plan in Fig. 3.

The cross-compound engine possesses a mechanical advantage not only over the simple expansive engine, but also over the tandem compound engine of equal horsepower. In a cross-compound engine the high-pressure cylinder *a* and low-pressure cylinder *b* are placed parallel to and alongside of each other, as shown in Fig. 3. Both cylinders are connected to a common crank-shaft *c*, which carries the flywheel *d* between the bearings and has cranks *e* and *f* at each end. These cranks are placed at right angles to each other, so that when one crank is on a dead center, the other is midway between the centers. In consequence of this there is a much more uniform turning moment on the crankshaft than in a tandem compound engine, and hence the

cross-compound engine will run steadier than the tandem compound, which runs steadier than the simple expansive engine. In comparing the steadiness of running, it is assumed that the flywheels are equal for engines of equal horsepower. From this the conclusion may be drawn that

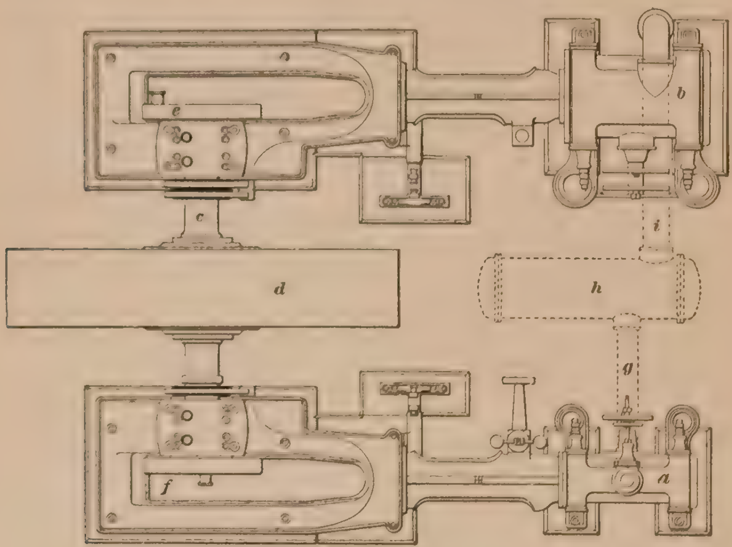


FIG. 3.

for equal degrees of steadiness the cross-compound will have the lightest flywheel, a heavier one is needed for the tandem compound, and the simple expansive engine needs the heaviest wheel.

In a cross-compound engine the exhaust from the high-pressure cylinder passes through the high-pressure exhaust pipe *g* (see Fig. 3) into the **receiver** *h* and thence through the receiver steam pipe *i* into the low-pressure steam chest. The exhaust from the low-pressure cylinder passes either into the atmosphere or into a condenser. The purpose of the receiver will be explained later.

CONSTRUCTION OF COMPOUND ENGINES.

CLASSIFICATION.

15. Types.—Compound engines may be divided into two general types, which are the *Woolf compound type* and the *receiver compound type*.

16. The Woolf Compound.—In the Woolf compound type the pistons of each cylinder commence and complete the stroke at the same time, and, consequently, the high-pressure cylinder can exhaust directly into the low-pressure cylinder. In Woolf compound engines either both pistons operate directly upon one crank, as in a tandem compound engine, or they operate upon two or more cranks on the same shaft, which are either in line or 180° apart. The absence of a receiver, or vessel destined to receive the high-pressure exhaust before it passes to the low-pressure cylinder, is the distinguishing feature of the Woolf compound engine. While most tandem compound engines are of the Woolf compound type, it must not be inferred that all of them belong to that type. A tandem compound engine does not need a receiver, as far as the steam distribution is concerned, but some of them, especially in very large sizes, are fitted with reheating receivers, where an additional amount of heat is supplied to the high-pressure exhaust in order to reduce the cylinder condensation in the low-pressure cylinder.

17. The Receiver Compound.—In the receiver compound type the high-pressure cylinder exhausts into a separate vessel, chiefly in order that the cranks may be placed at any desired angle other than 0° (in line with each other) or 180° apart. This is not feasible without the employment of a receiver, which, however, need not be a separate vessel, but may take the form of a very large exhaust pipe from the high-pressure to the low-pressure cylinder, or which may be formed by an exceptionally large low-pressure steam chest.

18. The reason why a receiver is needed when the cranks are at any other angle than 0° or 180° with each other may be explained as follows: Assume a cross-compound engine with cranks 90° apart and cutting off at ½ stroke in both cylinders. Let the high-pressure piston just have commenced its stroke. Then, the low-pressure piston will be just past its mid-stroke position and the low-pressure steam port is closed. But while the high-pressure piston is taking live steam on one side, it is exhausting on the other side, and there being no space for all the high-pressure exhaust steam to go into, the exhaust steam will be compressed to an extent depending on the volume of the low-pressure steam chest and the passage leading to it. This compression represents a waste of work, which is avoided by the use of a receiver.

19. Definitions.—The number of stages in which the expansion of the steam is carried on is denoted by calling the engine a *compound*, *triple-expansion*, *quadruple-expansion*, or *quintuple-expansion engine*. In a **compound engine** the steam is expanded in two separate stages, but not necessarily in two cylinders; in a **triple-expansion engine** the steam is expanded in three separate stages, but not necessarily in three cylinders, etc.

The expressions *compound*, *triple-expansion*, *quadruple-expansion*, etc. must not be taken to infer that the steam is expanded to twice, three times, four times, etc. its original volume, or that the engine has two, three, four, five, etc. cylinders. They refer solely to the number of different stages in which the steam is expanded. Thus, compound engines have been built with three cylinders and three cranks, there being one high-pressure cylinder and two low-pressure cylinders. In such an engine one-half of the exhaust steam from the high-pressure cylinder passes to each low-pressure cylinder. Such an engine is distinctly a *compound* engine, and may be more fully denoted as a *three-cylinder*, *three-crank*, *compound engine*.

20. All engines in which the expansion of the steam takes place in more than one stage are called **multiple-expansion engines,** to distinguish them from the single-cylinder or **simple expansive engine.** In giving the size of a multiple-expansion engine it is customary to give the sizes of their cylinders in inches, commencing with the smallest cylinder, and to write the stroke in inches last. Thus, a compound engine whose high-pressure cylinder is 11 inches in diameter, low-pressure cylinder 20 inches in diameter, and stroke 15 inches, would be designated as a $11'' \times 20'' \times 15''$ compound engine. In the same manner, the designation a $14'' \times 22'' \times 34'' \times 18''$ triple-expansion engine would mean that the diameters of the cylinders were 14 inches, 22 inches, and 34 inches, and that they had a common stroke of 18 inches.

21. In a triple-expansion engine there is at least one cylinder between the high-pressure and low-pressure cylinders. This cylinder is called the **intermediate-pressure cylinder.** In a quadruple-expansion engine there are at least two cylinders through which the steam must pass after leaving the high-pressure cylinder and before reaching the low-pressure cylinder. These are called the **first** and **second intermediate-pressure cylinders,** respectively.

CYLINDER ARRANGEMENTS.

22. It is customary to designate the cylinders and other component parts of compound and multiple-expansion engines as follows:

H. P. = high-pressure cylinder;
I. P. = intermediate-pressure cylinder;
1 I. P. = first intermediate-pressure cylinder;
2 I. P. = second intermediate-pressure cylinder;
L. P. = low-pressure cylinder.

In the description of the various cylinder arrangements, these designations have been adopted, and in Fig. 4 the

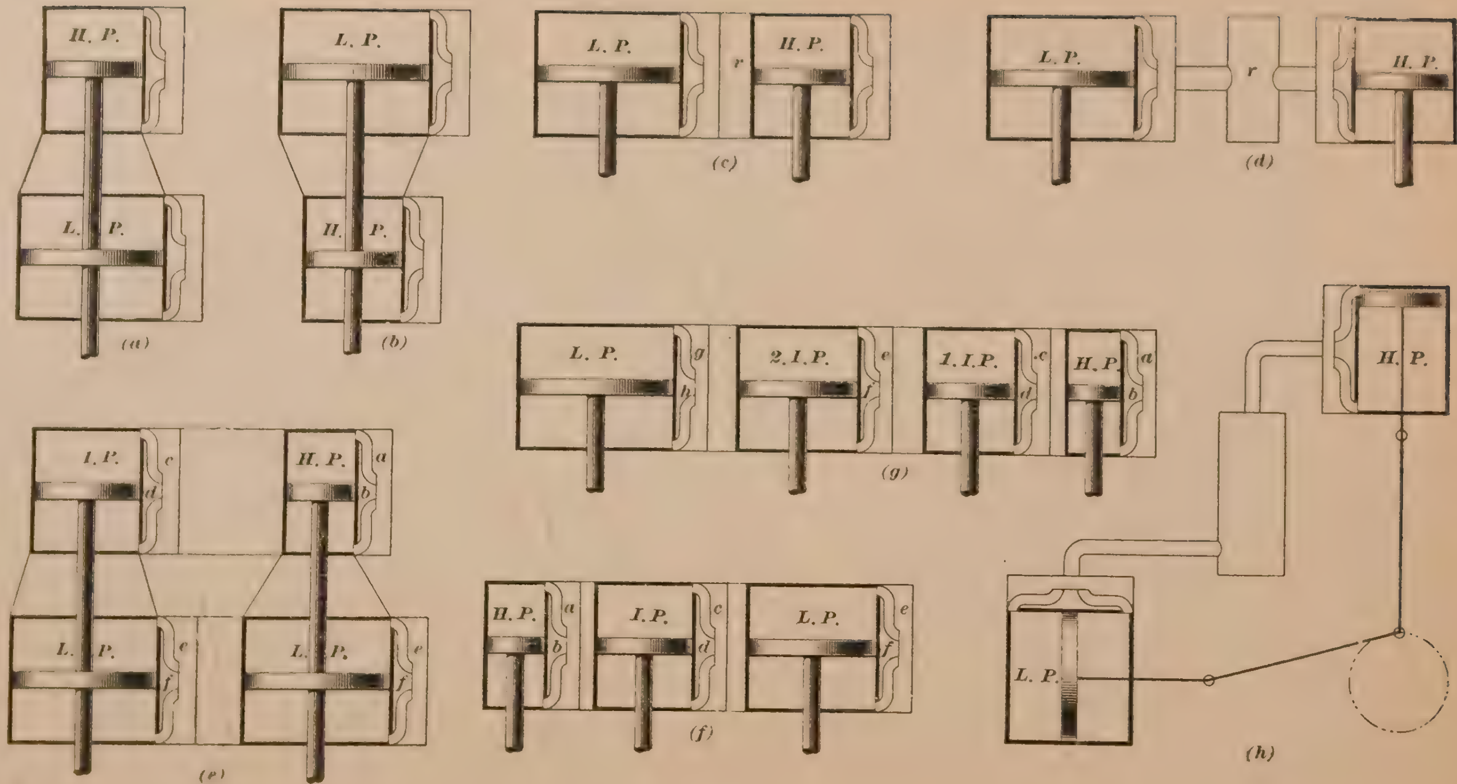

H. P.
L. P.
(a)
L. P.
H. P.
(b)
L. P.
r
H. P.
(c)
L. P.
r
H. P.
(d)
L. P.
g
h
2. I. P.
e
f
1. I. P.
c
d
H. P.
a
b
(g)
H. P.
L. P.
I. P.
c
d
H. P.
a
b
L. P.
e
f
L. P.
e
f
(e)
H. P.
a
b
I. P.
c
d
L. P.
e
f
(f)
(h)

FIG. 4.

cylinders have been lettered to correspond. In English books, *M. P.* designates the intermediate-pressure cylinder. The most common forms of compound, triple-expansion, and quadruple-expansion engines are illustrated in diagrammatic form in Fig. 4. The most common form of tandem compound engine is shown in Fig. 4 (*a*). Here the *L. P.* cylinder is next to the crank-shaft. In Fig. 4 (*b*) the low-pressure cylinder is furthest from the crank-shaft. Tandem compound engines of either design may be vertical or horizontal; when vertical they are often called **steeple compound engines.**

Fig. 4 (*c*) shows the cylinder arrangement of a cross-compound, high-speed engine. In these engines there are generally two flywheels placed outside of the cranks, the latter being at right angles with each other. The cylinders being close together, the receiver *r* is generally in the form of a jacket that ties the cylinders together. This arrangement of cylinders greatly lessens the floor space required. Fig. 4 (*d*) shows the cylinder arrangement of cross-compound engines of large power. It will be noticed that this is the same as shown in Fig. 3. Here the cylinders are separated in order to allow the flywheel to be placed between the cranks. The receiver *r* is generally a separate vessel. Cross-compounds may be either vertical or horizontal.

Fig. 4 (*e*) shows the cylinder arrangement of a two-crank, four-cylinder, triple-expansion engine. Steam enters the *H. P.* cylinder from the steam chest *a*; it exhausts through *b* into the intermediate receiver *c*, whence it passes into the *I. P.* cylinder. The latter exhausts through *d* into the low-pressure receiver *e e*, common to both *L. P.* cylinders Through *f f* the exhaust steam from the *L. P.* cylinders passes into the condenser. The cranks are placed 90° apart. For large powers the cylinders may be separated, similar to the cross-compound engine shown in Fig. 4 (*d*). In that case the low-pressure receiver would most likely be a separate vessel. Fig. 4 (*f*) shows the cylinder arrangement of a three-crank, three-cylinder, triple-expansion engine with the cylinders placed close together. In this

case the flywheel is placed on one end of the crank-shaft. The cranks are usually placed 120° apart. The steam from the boiler enters the *H. P.* steam chest *a* and is exhausted from the *H. P.* cylinder through *b* into the intermediate receiver and passes thence into the *I. P.* steam chest *c*. The steam exhausts from the *I. P.* cylinder through *d* into the low-pressure receiver, whence it passes into the *L. P.* steam chest *e*, and finally exhausts through *f* into the atmosphere or condenser.

Fig. 4 (*g*) illustrates the cylinder arrangement of a four-crank, four-cylinder, quadruple-expansion engine. Steam enters the *H. P.* cylinder from the steam chest *a*; it exhausts through *b* into the first intermediate receiver *c*, whence it passes into the *1 I. P.* cylinder. The exhaust from the latter passes through *d* into the second intermediate receiver *e*, and thence into the *2 I. P.* cylinder. This cylinder exhausts through *f* into the low-pressure receiver *g*, whence the steam passes into the *L. P.* cylinder and exhausts through *h* into the condenser. The cranks are placed 90° apart.

Fig. 4 (*h*) shows a cylinder arrangement for a compound engine that is beginning to find much favor with designers of late for large powers. The two cylinders are placed at right angles to each other, and both pistons are connected to the same crankpin. Sometimes two sets of such engines are placed alongside of each other, with the flywheel between them. A compound engine arranged in this manner will occupy much less floor space than a cross-compound engine, and will run just as steadily as the latter, owing to the uniformity of the turning moment on the crank-shaft.

EXAMPLES OF COMPOUND ENGINES.

23. Tandem Compound.—An elevation of a tandem compound, high-speed engine is shown in Fig. 5. In the illustration, *a* is the high-pressure cylinder, which in high-speed tandem compound engines is generally placed behind the low-pressure cylinder *b*. In large medium-speed engines the high-pressure cylinder is often placed nearest the

crank-shaft, it being claimed that it is then easier to remove the pistons and to examine the cylinders in case of repairs. When the high-pressure cylinder is behind, as in Fig. 5, it must generally be removed entirely in order to get at the

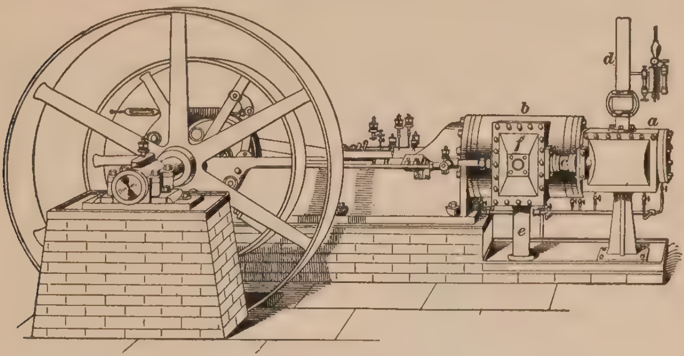


FIG. 5.

inside of the low-pressure cylinder. Steam is conducted to the high-pressure steam chest by the steam pipe *d*; after the steam has expanded in *a* it is discharged through the

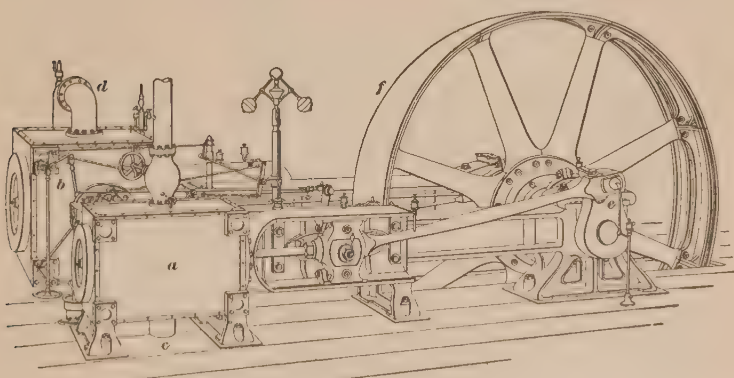


FIG. 6.

connecting pipe *c* into the steam chest *f* of the low-pressure cylinder *b*, and is finally exhausted into the condenser or atmosphere through the exhaust pipe *e*. As in nearly all

high-speed engines, a shaft governor is used. The engine is of the side-crank type; that is, the crank is on the end of the crank-shaft. Many tandem compound engines are built, however, with a center crank, which means that the crank is in the center of the crank-shaft. In that case there are usually two flywheels, one on each side.

24. Horizontal Cross-Compound.—A perspective view of a cross-compound, horizontal engine, with Corliss valve

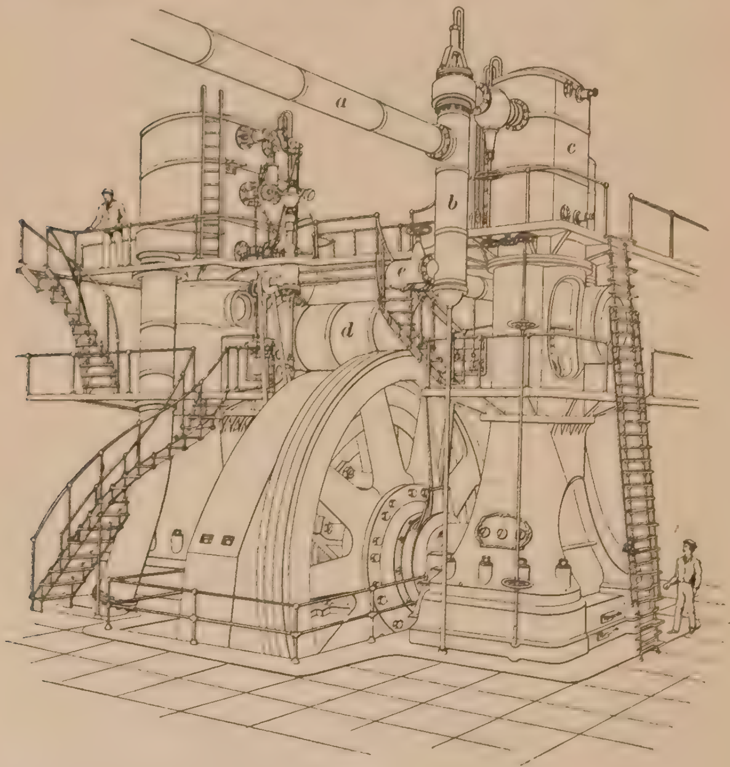


Fig. 7.

gear, is given in Fig. 6. In the illustration, *a* is the high-pressure cylinder and *b* the low-pressure cylinder. The receiver between the two cylinders is placed beneath the floor, the pipe *c* leading to it from the high-pressure

cylinder and the pipe *d* leading from it to the low-pressure steam chest. The cranks are placed 90° apart; owing to the fact that the high-pressure crank *e* is on its upper quarter, the low-pressure crank is hidden by the frame. The flywheel *f* serves as a belt wheel and is placed between the cranks.

25. Vertical Cross-Compound.—Fig. 7 shows a perspective view of a vertical cross-compound engine direct-connected to a dynamo and having Corliss valve gear. The steam is led to the engine through the main steam pipe *a*, and before passing into the high-pressure steam chest passes into a separator *b*, which removes the entrained water. The exhaust steam from the high-pressure cylinder *c* passes into the reheating receiver *d*, where the exhaust steam is superheated by live steam taken from the bottom of the separator through the pipe *e*. The dynamo is placed between the bearings, its armature being keyed directly to the crank-shaft.

The particular engine shown is about 4,500 horsepower; it has cylinders 46 and 86 inches diameter, a stroke of 60 inches, and a speed of 75 revolutions per minute.

26. Duplex Vertical and Horizontal Compound.—Fig. 8 is a perspective view of an 8,000 horsepower duplex compound engine with the two cylinders of each engine at right angles to each other. The two engines are duplicates of each other and are so arranged that either engine can be uncoupled. As shown, the pistons of the high-pressure cylinders *a*, *a* and low-pressure cylinders *b*, *b* of each engine are connected to a common crankpin, as *c*. The cranks of this engine are placed 45° apart, so that the crank-shaft receives eight impulses during each revolution, which gives such a uniform turning effect that the flywheel is dispensed with, its place, to some degree, being taken by the revolving field *d* of the dynamo. The steam coming from the high-pressure cylinders passes through reheating receivers *e*, *e*. The engine has high-pressure cylinders 44 inches in diameter,

the low-pressure cylinders are 88 inches in diameter, the stroke is 60 inches, and the speed is 75 revolutions per

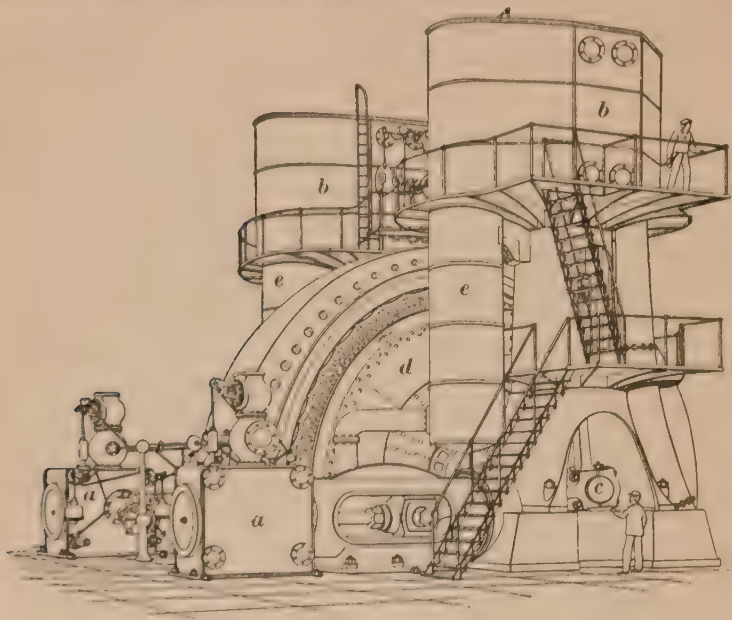


FIG. 8.

minute. The engine can be worked up to 12,000 horsepower.

27. Triangular Connecting-Rod Engine.—Fig. 9 is an end view of one side of a quadruple-expansion, two-crank engine having the so-called **triangular connecting-rod** patented by John Musgrave & Sons, Bolton, England. The engine is of the vertical type and is fitted with Corliss valve gear. The four cylinders are disposed in pairs on each side of the flywheel or rope drum, the high-pressure and first intermediate cylinders being on one side of the drum and the second intermediate and low-pressure cylinders are on the other side. The crossheads of each pair of cylinders are connected by means of a pair of links *a*, *a* and a triangular connecting-rod *b* to a single crank *c*; the two

cranks of the engine are opposite each other, so that the weights of the two sets of reciprocating parts mutually balance each other.

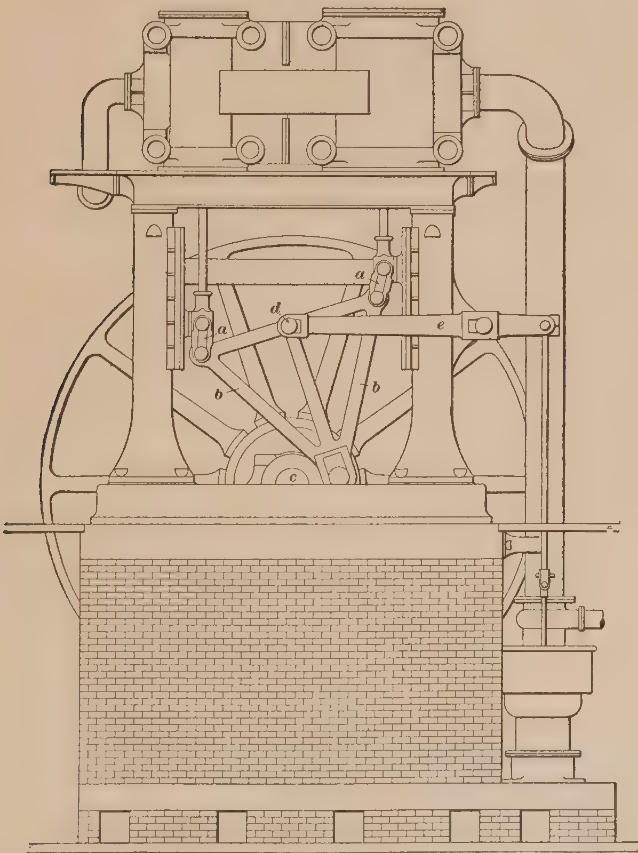


FIG. 9.

The action of the triangular connecting-rod is such that there are no dead centers to the engine, as although the crossheads of both cylinders are connected to one crank, they are never at the ends of their respective strokes at the same time, and the turning effort is the same as if they

were connected to cranks set nearly at right angles to each other; that is, when one piston is at the end of its stroke the other is nearly in its central position and has a very effective leverage to turn the crank. But a further and very important effect of this arrangement is that the strains on the crank are gradually changed *around* the crankpin from one side to the other; they are never suddenly reversed, as is the case with an ordinary engine, and in consequence these engines may be run at very high speeds without any noticeable jar or vibration. The triangular connecting-rod mentioned vibrates on a pin *d* in the ends of a pair of levers, as *e*, which latter swing on a fixed center just outside of the right-hand frame. Extensions of these levers outside of the frames are made use of to work the air pump. The motion of the ends of the triangular connecting-rod, to which the crossheads are connected, is a vibrating one, due both to the arc formed by the swinging lever and to the circular path of the crankpin. In consequence of this combined movement, these ends of the rod move up and down in nearly vertical lines, so that the side pressure on the guides is very small, in fact it is much less than in an ordinary engine with a connecting-rod of the usual length.

THE RECEIVER.

28. In practice the receiver is often given such dimensions that it has a volume equal to that of the low-pressure cylinder. When its volume is that large, the pressure within it varies but little, so long as the weight of steam supplied by the high-pressure cylinder in one revolution of the engine is equal to that drawn out by the low-pressure cylinder. Assuming that the pressure in the receiver is constant and neglecting the resistance of the passages connected with the cylinders, the back pressure in the high-pressure cylinder will be the same as the initial pressure in the low-pressure cylinder. In each cylinder, then, the steam is admitted, cut off, and expanded, just as if there were two independent

engines, each of which uses the same weight of steam per revolution.

29. The pressure existing in the receiver can and does exert a marked influence on the engine, both in regard to its operation as a machine and as a heat engine. The receiver being a vessel having a constant volume, the pressure within it depends on two things which are the rate at which steam is discharged into it and the rate at which it is drawn out. When the volume of steam drawn out is greater than the volume of steam in the high-pressure cylinder at the moment release begins in that cylinder, it will be apparent that the exhaust steam rushing into the receiver must expand to the volume drawn from the receiver. In expanding, the pressure of the exhaust steam drops, so that the pressure in the receiver, and of course the back pressure on the high-pressure piston, is less than the pressure at the time of release. This difference in pressure at the time of release and in the receiver is termed **drop.**

30. When saturated steam is allowed to expand without doing any work, as is the case when the receiver pressure is less than the pressure at the point of release, the steam becomes superheated to an extent depending on the drop. The effect of this superheating is to partially or entirely evaporate any water in the steam, thus making it drier and tending to reduce the initial condensation in the low-pressure cylinder. It may be incidentally remarked here that the expanding of steam without doing any work is termed **free expansion.**

31. The most economical receiver pressure for a given engine can only be found by experiment, regulating the pressure and noting the effect upon the steam consumption. The receiver pressure is regulated in several ways. First, by making the cut-off later in the high-pressure cylinder and leaving the low-pressure cut-off as before. Since there is now a larger volume of steam entering the receiver, the

receiver pressure will be higher. The initial pressure will also be higher in the low-pressure cylinder, and the cut-off being the same as before, the mean effective pressure will be greater. Since the mean effective pressure in the high-pressure cylinder is also greater, it is seen that the effects of a later cut-off in the high-pressure cylinder is an increase of the power of the engine.

32. Making the cut-off earlier in the high-pressure cylinder means a lower pressure at release and a lower receiver pressure; consequently, the mean effective pressures in the two cylinders, and hence the power of the engine, are decreased by this means.

33. The amount of drop can only be regulated by means of the low-pressure cut-off. Thus, leaving the cut-off of the high-pressure cylinder as before, the pressure at release will be the same. But if the cut-off is made later in the low-pressure cylinder, a larger volume of steam will be drawn from the receiver, and, consequently, the receiver pressure will be less and the drop more. Conversely, if the low-pressure cut-off is made earlier, there will be less steam drawn from the receiver, and hence the receiver pressure will be higher and the drop less.

GOVERNING A COMPOUND ENGINE.

34. Obviously a change in the receiver pressure due to a manipulation of the low-pressure cut-off will change materially the amount of work done in each cylinder, although it will leave the total work done by the two cylinders the same as before. Thus, if the receiver pressure is lowered by making the low-pressure cut-off later, the back pressure obviously will be less in the high-pressure cylinder, and hence the mean effective pressure will be more, so that the high-pressure cylinder will now be doing more work. Since the total amount of work done remains the same as before, it follows that the low-pressure cylinder, in spite of its later cut-off, will now be doing less work. Conversely,

if the receiver pressure is raised by making the low-pressure cut-off earlier, less work will be done in the high-pressure cylinder and more in the low-pressure cylinder.

35. From the foregoing statements it will be plain that in order to change the power of the engine, the high-pressure cut-off must be changed, while in order to change the distribution of work in the two cylinders, the low-pressure cut-off must be changed. This will explain why the governing mechanism of many compound engines operates upon the high-pressure cut-off, which manner of governing is open to one serious objection, however. When the governor makes the high-pressure cut-off later in responding to an increase of load, the larger share of the total work will be done in the low-pressure cylinder; conversely, if the governor makes the cut-off in the high-pressure cylinder earlier, the larger share of the work will be done in the high-pressure cylinder, and at very early cut-offs the low-pressure cylinder may then actually be a drag on the engine.

36. In order to overcome the objectionable features of controlling only the high-pressure cut-off, many compound engines have the governor operate both the high-pressure and the low-pressure cut-offs. In that case the governor operates the high-pressure cut-off in order to change the power of the engine and the low-pressure cut-off in order to equalize the work done in the two cylinders. That is, when the governor makes the cut-off later in the high-pressure cylinder, it also makes it later in the low-pressure cylinder; conversely, when the load is lighter, the governor makes the cut-off earlier in both cylinders.

RECEIVER FITTINGS.

37. Receivers should always be fitted with gauges registering from zero to the maximum pressure in one direction and from zero to 30 inches of vacuum in the other direction, i. e., with compound gauges. The receiver should also be fitted with pop safety valves of large capacity, so that in

case the low-pressure cut-off should be set too early, or, if a releasing gear is used, if the gear should fail to open the low-pressure admission valves, the steam would be relieved. Otherwise, the high-pressure cylinder, aided by the momentum of the flywheel, would pump the receiver full of steam and cause it to explode.

SETTING THE VALVES OF MULTIPLE-EXPANSION ENGINES.

38. In setting the valves of horizontal, compound, and triple-expansion engines, considerable more lead should be given in the low-pressure cylinder. The lead for a 48-inch, low-pressure, condensing cylinder could be from $\frac{1}{4}$ to $\frac{5}{16}$ inch, and if the back pressure is very low the lead may be somewhat larger. The valves of each cylinder of a compound and triple-expansion engine must be set in relation to the crank on which that cylinder is doing its work. For vertical engines it is customary to give a little more lead on the bottom than on the top, for the reason that the force required to accelerate the reciprocating parts is greater on the up stroke than on the down stroke, due to the weight of the reciprocating parts. In many of the valve gears now in use, the action of the valve is so quick at the instant of opening that many engineers now measure the lead not in the amount that the valve is open when the engine is on the center, but by the position of the crank when the valve begins to open. The practice with some engineers is to have the valve begin to open when the crank is yet 15° from the dead center. This angle, however, may be regarded as a limit rather than as an average. Perhaps 7° to 10° would be the average.

EQUALIZING WORK DONE IN THE CYLINDERS.

39. The question of whether or not it is necessary that both cylinders of a compound engine should do an exactly equal amount of work is at present in a somewhat unsettled state as far as cross-compound engines are concerned. From

the builder's standpoint it is more desirable to have the initial loads on the piston rods, crossheads, connecting-rods, and crankpins equal in order that these parts may be duplicates and yet of equal strength. The initial load is given by multiplying the area of the piston by the initial steam pressure. Obviously, it is independent of the cut-off, that is, the initial load is the same whether the cut-off is at $\frac{1}{10}$ or $\frac{7}{10}$ of the stroke, so that it can be readily seen that there is no relation between the initial load and the work done in the cylinder. Consequently, it is entirely possible with an equal division of work to have the maximum stresses on the parts of one side of the engine far in excess of those on the parts of the other side. For this reason it is urged by many engineers that in cross-compound engines, where the parts previously named are in duplicate, to so distribute the work that the initial loads will be approximately equal.

40. Opinions differ as to whether a cross-compound engine will run more satisfactory when the work is equally divided, and a decision one way or the other is best based on an actual trial. The mechanical operation of the engine is here referred to, not its economy.

41. In tandem compounds the work is generally evenly distributed between the cylinders, but even quite a perceptible variation seems to have no harmful effect, as far as the quiet running of the engine is concerned.

42. In practice it is probably the best plan with a given engine, when the design permits it, to distribute the work between the cylinders so as to obtain the smallest possible water consumption compatible with steady and satisfactory running. This object in any case can only be attained by repeated trials, changing the receiver pressures in order to change the work done in each cylinder and noting the water consumption after each change. In other words, it is the best plan to try to secure the highest economy obtainable in conjunction with a satisfactory mechanical operation

43. In many compound engines, especially in engines of the high-speed type, the distribution of work in the two cylinders has been determined upon by the builder and cannot be changed by any means at the command of the attendant. The valves of both cylinders are generally under the control of the governor, and the angle of advance then being a fixed quantity, which cannot readily be changed, it follows that all the attendant can do is to keep the valves centrally set and the engine in first-class running condition. There is little or no chance to improve the economy, i. e., to lower the steam consumption, at least, as far as the engine itself is concerned.

THE STEAM JACKET.

44. When the cylinder of an engine is surrounded with a casing and the space between the cylinder and casing is filled with live steam during running, the cylinder is said to be **steam-jacketed.** In comparatively slow-speed engines, say up to 150 revolutions per minute, using saturated steam, a properly applied and properly cared for steam jacket will generally produce a distinct gain in economy, while one that is improperly applied or cared for will prove a loss and sometimes a very serious one. If possible it is always better to use steam of a higher temperature in the jacket than that which enters the cylinder. In a compound engine a jacket may be applied to the high-pressure cylinder and the receiver, and in a triple-expansion engine to the first and second cylinders and both receivers, but in neither case need it be applied to the low-pressure cylinder. The object of the steam jacket is to prevent condensation taking place in the cylinder; if the steam that runs the engine can be superheated sufficiently to prevent initial condensation, there will then be no need of a steam jacket, and the engine will be more economical without a steam jacket than with one.

45. In order to gain a clear understanding of why a steam jacket under proper conditions will improve the economy,

the action of the steam that actually takes place in a steam cylinder after the engine is fairly started and when using ordinary saturated steam should be understood. This action in an unjacketed cylinder is as follows:

Steam enters at the beginning of the stroke, and as it is hotter than the metal with which it comes in contact, a portion of the entering steam is condensed in heating up the cylinder head, piston, and cylinder to the temperature of the steam. As the piston moves forwards, uncovering fresh surfaces of the cylinder, the condensation continues until after the admission of steam ceases—i. e., until the point of cut-off is reached. The steam then commences to expand and correspondingly lowers in pressure and temperature, and unless the steam is cut off very early in the stroke, no further condensation occurs after the cut-off has taken place (although for some distance further the piston uncovers fresh surfaces of the cylinder that are cooler than the steam) for the reason that, as soon as the temperature of the steam falls through expansion, the head, piston, and walls of the cylinder already heated begin to give up their heat to the steam and prevent further condensation taking place. As the expansion is carried still farther and the temperature of the steam correspondingly lowered, a portion of the water previously condensed is reevaporated into steam by the heat given up by the piston, cylinder, and head, raising the terminal pressure in the cylinder. When the piston arrives at the end of the stroke, the exhaust valve opens and the pressure and temperature of the steam lowers immediately, the remainder of the water previously condensed then being reevaporated by the heat extracted from the cylinder walls, piston, and head, and passes out with the exhaust steam. The exhaust steam is much colder than the metal, and during the whole of the return stroke is absorbing heat from the piston, cylinder walls, and head, lowering their temperature, which has to be raised again by the steam entering for the next forward stroke. The action of the heat above described takes place very quickly and affects the metal of the cylinder and head to a slight depth only, as there is not time for it to

penetrate very deeply; but it affects it at each stroke of the engine, and a small loss each stroke aggregates a very large one during the day's run. This action is also greatly increased by water held in suspension or entrained in the steam; it is therefore important that dry steam only should be used.

46. The action that takes place in a steam-jacketed cylinder is quite different, as the following description will indicate: Suppose the jackets to contain steam at the same temperature as the steam entering the cylinder—this being the usual case with the high-pressure cylinder of a compound or triple-expansion engine—and also suppose the jackets to be so arranged that neither air nor water can collect and interfere with their usefulness. At the beginning of the stroke the cylinder and head will be at the same temperature as the entering steam, and none of the entering steam will be condensed by them, as in the case of an unjacketed cylinder; the piston itself will be in the same condition as before, and a portion of the steam will be condensed and reevaporated by it. After the admission of steam ceases and the temperature of the steam inside the cylinders becomes less through expansion, heat is transferred from the cylinder walls to this steam, reevaporating the small amount of water condensed by the piston. The walls, in turn, absorb heat from the steam in the jacket, condensing some steam in the jacket, but maintaining the cylinder at nearly an even temperature. When the exhaust valve opens and the pressure and temperature of the steam in the cylinder become lower, there is no water to reevaporate; but as the steam inside the cylinder is then much cooler than the walls of the cylinder, it absorbs heat from them during the whole of the return stroke, which heat, in turn, is absorbed from the steam in the jacket, and this heat is practically wasted. The actual gain effected by the use of a steam jacket on a cylinder is the difference between the saving due to the prevention of condensation in the cylinder at the beginning of the stroke and the loss by heating the exhaust steam during the return stroke; and this gain may in many cases be very slight.

REHEATERS.

47. A **reheater** is a modification of the receiver commonly used in multiple-expansion engines and consists essentially of a receiver containing coils or nests of small pipes, through which high-pressure steam circulates and which are so arranged that the working steam must circulate around them thoroughly and become **reheated**, from which fact the apparatus receives its name. Formerly, receivers were simply reservoirs placed between the cylinders. The next step was to provide these receivers with steam jackets, and from noticing the economy obtained from the steam-jacketed receiver, the reheater was developed. It is claimed by advocates of the reheater that a gain of 10 per cent. can be realized by their use under favorable conditions. This, however, depends entirely on circumstances, and a reheater, instead of causing a gain, may cause a positive loss. Furthermore, it can seldom or never be predicted beforehand whether a reheater will be a gain or a loss. However, it can always be determined afterwards whether the action of the reheater is profitable. If a thermometer placed in the low-pressure exhaust shows that the temperature of the exhaust steam is higher than that corresponding to its pressure, the reheater is wasting heat, and the reheating should be abandoned. In proportioning the reheating surface, 40 square feet of surface are generally allowed for every cubic foot of steam swept into the reheater by the high-pressure piston during exhaust.

48. Fig. 10 shows a common form of a reheater. It consists of a cast-iron or wrought-iron shell *a*, having at one end a tube plate *b* secured between the shell *a* and cover *c*. A large number of wrought-iron boiler tubes, as *d*, *d*, are expanded into this tube plate. These boiler tubes are expanded at the other end into the tube plate *e*, which is bolted to the head *f*. It will be noticed that the head *e* and tube plate *f* form a vessel that is independent of the shell of the receiver, so that the tubes are perfectly free to expand

or contract. The right-hand end of the reheater is closed by the cover *g* through which the steam pipe *h* and air drain pipe *i* pass. Stuffingboxes are used to make a steam-tight joint where these pipes pass through the head. An automatic air valve is generally attached to the pipe *i*, through which the air is discharged when starting up and which closes as soon as steam reaches it. The pipe *h* serves as a drain and is led to a trap. The drain for the shell is attached at *k* and is also led to a trap.

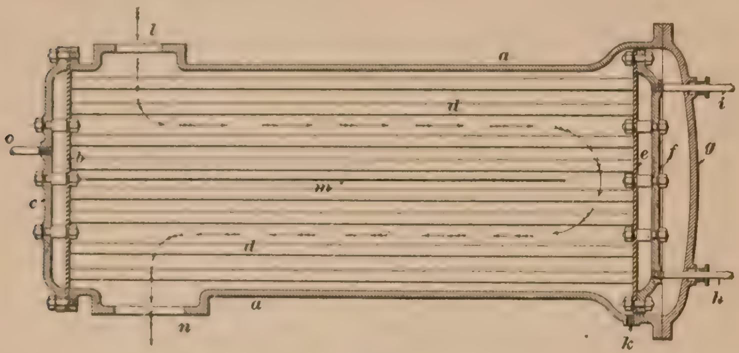


FIG. 10.

The exhaust from the high-pressure cylinder enters at *l* and is compelled by the baffle plate *m* to flow in the path indicated by the arrows over the tubes *d*, *d*, and finally passes through the opening *n* to the low-pressure steam chest. Live steam at boiler pressure is admitted to the tubes through the pipe *o* and leaves through *h*.

49. For very large engines reheaters are often constructed similar to tubular boilers, the tubes being expanded into rigid tube-sheets riveted to the shell. The working steam, then, generally passes through the tubes, while the steam used for reheating circulates outside of them. With such a reheater it is a good plan to take the live steam for the reheater from a connection placed between the throttle and the engine. If this is done, steam will be admitted to both sides of the reheater as soon as the engine is started,

and, in consequence, its expansion will be more uniform, which renders it less liable to leakage around the tube ends.

50. A reheating receiver should, in general, never be used when the engine uses superheated steam, for the reason that the steam entering the receiver usually contains enough superheat to prevent initial condensation, and the use of a reheating receiver under these conditions will result in a direct waste of heat.

CALCULATIONS PERTAINING TO COMPOUND ENGINES.

RATIO OF EXPANSION.

RATIO OF EXPANSION BY VOLUME.

51. The ratio of expansion of a multiple-expansion engine is generally given as the ratio between the volume of steam in the low-pressure cylinder at low-pressure release, that is, at the time the low-pressure exhaust valve opens, and the volume of steam admitted to the high-pressure cylinder. The ratio of expansion is independent of the low-pressure cut-off or volume of the receiver.

52. The ratio of expansion of a multiple-expansion engine may be found in two different ways. In the formulas corresponding to the following rules:

E = total ratio of expansion;
v = volume of high-pressure cylinder up to the point of cut-off, including clearance;
V = volume of low-pressure cylinder at release, including the clearance;
e = ratio of expansion in the high-pressure cylinder;
v_1 = volume swept through by high-pressure piston, clearance included.

Rule 1.—*To find the total ratio of expansion, divide the volume of the low-pressure cylinder up to the point of release and including clearance by the volume of the high-pressure cylinder up to the point of cut-off and including clearance.*

Or,
$$E = \frac{V}{v}.$$

Rule 2.—*To find the total ratio of expansion, multiply the ratio of expansion of the high-pressure cylinder by the volume of the low-pressure cylinder up to the point of release and divide by the volume swept through by the high-pressure piston, clearance included.*

Or,
$$E = \frac{e\,V}{v_1}.$$

When applying rules **1** and **2** to a triple-expansion or quadruple-expansion engine, the intermediate cylinders are to be neglected entirely; that is, consider only the first and last cylinders. The ratio of expansion calculated by rules **1** and **2** is known as the **ratio of expansion by volume**, and when the expression "ratio of expansion" is used without any particular qualification, it is always understood to mean *by volume*.

EXAMPLE 1.—The volume of the high-pressure cylinder of a triple-expansion engine, up to the point of cut-off, is 650 cubic inches; the volume of the low-pressure cylinder is 13,650 cubic inches. What is the total ratio of expansion?

SOLUTION.—Applying rule **1**, we have

$$E = \tfrac{13650}{650} = 21. \quad \text{Ans.}$$

EXAMPLE 2.—The volume of the low-pressure cylinder of a compound engine is 16,548 cubic inches. The volume swept through by the high-pressure piston, including clearance, is 4,234 cubic inches. The ratio of expansion of the high-pressure cylinder being 2.6 what is the total ratio of expansion?

SOLUTION.—Applying rule **2**, we have

$$E = \frac{2.6 \times 16{,}548}{4{,}234} = 10.16. \quad \text{Ans.}$$

RATIO OF EXPANSION BY PRESSURE.

53. The ratio of expansion of a multiple-expansion engine is sometimes given as the ratio between the absolute pressure at the point of cut-off in the high-pressure cylinder and the absolute pressure in the low-pressure cylinder at the point of release. The ratio of expansion by pressure is, for the same engine, always greater than the ratio of expansion by volume, owing to drop in the receiver. The absolute pressures are taken by measurement from indicator diagrams.

HORSEPOWER OF COMPOUND ENGINES.

CALCULATING THE INDICATED HORSEPOWER.

54. The indicated horsepower of a compound or triple-expansion engine is calculated from the indicator diagrams in exactly the same manner as with any simple engine, considering each cylinder as a simple engine and adding the horsepowers of the engines together. In taking the indicator cards from a compound engine, the precaution of taking the cards simultaneously from all cylinders must be observed, especially when the engine runs under a variable load, since otherwise an entirely wrong distribution of power may be shown, and there may also be a great variation between the indicated horsepower really existing and that calculated from diagrams taken at different times.

55. The indicated horsepower of compound engines is sometimes found by referring the mean effective pressure of the high-pressure cylinder to the low-pressure cylinder and calculating the horsepower of the engine on the assumption that all the work is done in the low-pressure cylinder. To do this, the mean effective pressures of the two cylinders are found from indicator diagrams; the mean effective pressure of the high-pressure cylinder is then divided by the ratio of the volume of the low-pressure cylinder to that of

the high-pressure cylinder, and the quotient is added to the mean effective pressure of the low-pressure cylinder. The sum is then taken as the mean effective pressure of the engine, and the area of the low-pressure piston as the piston area; with these data, the length of stroke and the number of strokes, the horsepower is computed as for any simple engine. In the case of a triple-expansion engine, the mean effective pressures of the high-pressure and intermediate cylinders are referred to the low-pressure cylinder and added to its mean effective pressure. While this method shortens the labor of computing the horsepower, it obviously does not show the distribution of work between the cylinders, and for this reason is going rapidly out of use.

ESTIMATING THE INDICATED HORSEPOWER.

56. The horsepower of a multiple-expansion engine cannot be estimated with any great degree of accuracy, and in practice it would be considered a satisfactory result if the indicated horsepower and estimated horsepower agree within 10 per cent.

57. The difficulty of determining the horsepower is due to the fact that the mean effective pressure cannot be estimated very closely, and for this reason it is better to take the mean effective pressure from diagrams whenever this is feasible.

58. The horsepower of a multiple-expansion engine is estimated by estimating the horsepower of a simple expansive engine having the same dimensions as the low-pressure cylinder, the same piston speed, the same initial pressure, and the same total ratio of expansion. The problem then resolves itself into finding the probable mean effective pressure of this equivalent simple expansive engine.

59. To estimate the probable mean effective pressure, it is necessary to assume an initial and a terminal pressure in

order to determine the probable ratio of expansion by pressure. With a steam pipe of ample size and the throttle wide open, the initial pressure in the high-pressure cylinder will probably average 5 pounds per square inch below the boiler pressure. The terminal pressure, i. e., the pressure at the point of release in the low-pressure cylinder, may be assumed to average 9 pounds absolute in condensing engines and 19 pounds absolute in non-condensing engines. Then, the total ratio of expansion, by pressure, will be the absolute initial pressure divided by the absolute terminal pressure. This determination of one of the factors (the total ratio of expansion), being based on assumptions that may differ considerably from the conditions actually existing, will serve to show why the determination of the mean effective pressure is only approximate at best, and that, hence, the horsepower of the engine cannot be estimated with any great degree of accuracy.

Rule 3.—*To estimate the probable mean effective pressure of a simple expansive engine equivalent in power to a given multiple-expansion engine, multiply the estimated initial absolute pressure in the high-pressure cylinder by the factor opposite the ratio of expansion in Table I. Subtract the estimated back pressure and multiply the remainder by the factor corresponding to the type of engine and given in Table II.*

Or, $$p_m = (p_i c - p_b) \times f,$$

where p_m = mean effective pressure;
p_i = initial absolute pressure in the high-pressure cylinder;
c = factor taken from Table I;
p_b = absolute back pressure;
f = factor taken from Table II.

The absolute back pressure may be estimated at 17 pounds for non-condensing engines and 3 pounds for condensing engines. The values here given are average values and may vary somewhat from the true values that may actually

TABLE I.

FACTORS CORRESPONDING TO RATIOS OF EXPANSION.

Ratio of Expansion.	Factor.	Ratio of Expansion.	Factor.	Ratio of Expansion.	Factor.	Ratio of Expansion.	Factor.
1.32	.966	2.86	.717	7.00	.421	16.0	.236
1.48	.940	3.00	.699	8.00	.385	17.0	.226
1.54	.929	3.33	.661	8.50	.369	18.0	.216
1.60	.918	3.63	.631	9.00	.355	19.0	.207
1.66	.906	4.00	.596	10.00	.330	20.0	.199
1.82	.878	4.44	.561	11.00	.309	21.0	.192
2.00	.846	5.00	.522	12.00	.290	22.0	.186
2.22	.809	5.71	.480	13.00	.274	24.0	.174
2.50	.766	6.00	.465	14.00	.260	26.0	.164
2.66	.744	6.60	.434	15.00	.247	28.0	.155

TABLE II.

FACTORS CORRESPONDING TO TYPE OF ENGINE.

Type of Engine.	Ratio of Expansion.	Factor.
Compound, non-condensing Corliss.....	6–12	.85
Compound, condensing Corliss.........	12–18	.80
Compound, non-condensing slide-valve. .	6–12	.75
Compound, condensing slide-valve......	12–18	.70
Triple-expansion, condensing Corliss....	18–27	.72
Triple-expansion, condensing slide-valve	18–27	.60

obtain with a given engine, and thus another source of error is introduced in the estimation of the mean effective pressure.

60. The mean effective pressure having been estimated by rule **3,** the horsepower is calculated by multiplying together the estimated mean effective pressure, the length of stroke in feet, the area of the low-pressure piston in square inches, and the number of strokes per minute, and dividing the product by 33,000. It will be noticed that this is the same rule by which the horsepower of any simple engine is calculated.

EXAMPLE.—Estimate the probable horsepower of a triple-expansion engine having cylinder diameters of 22, 35, and 56 inches; a common stroke of 36 inches; a speed of 140 revolutions per minute; and a boiler pressure of 175 pounds. The engine is condensing and fitted with piston valves throughout.

SOLUTION.—The absolute initial pressure may be estimated at $(175 + 14.7) - 5 = 184.7$, say 185 pounds per square inch. The terminal pressure, by Art. **59,** may be taken as 9 pounds. Then the ratio of expansion by pressure is $\frac{185}{9} = 20.55$. By Table I, the factor for a ratio of expansion of 20 is .199, and for 21 is .192. Since 20.55 is about midway between 20 and 21, the factor to be used may safely be assumed to be midway between .199 and .192, say .195. The back pressure, by Art. **59,** may be estimated at 3 pounds. Since the engine is fitted with piston valves, which are the equivalent of slide valves, the factor to be taken from Table II is .6.

Applying rule **3,** we get

$$p_m = (185 \times .195 - 3) \times .6 = 19.8 \text{ pounds per square inch.}$$

Then, the probable indicated horsepower

$$= \frac{19.8 \times \frac{36}{12} \times 56^2 \times .7854 \times 2 \times 140}{33,000} = 1,241.35, \text{ say } 1,250. \quad \text{Ans.}$$

CYLINDER VOLUMES.

61. The average ratios of cylinder volumes of compound and triple-expansion engines with different steam pressures are given in Table III.

TABLE III.

AVERAGE RATIOS OF CYLINDER VOLUMES.

Initial Steam Pressure, Gauge.	High-Pressure Cylinder.	Inter-mediate Cylinder.	Low-Pressure Cylinder.	Remarks.
100	1		2.60	Non-condensing
100	1		3.60	Condensing
110	1		3.80	Condensing
120	1		4.00	Condensing
130	1		4.15	Condensing
140	1		4.30	Condensing
150	1		4.45	Condensing
160	1		4.60	Condensing
160	1	3.00	7.00	Condensing
185	1	3.33	7.80	Condensing

ENGINE MANAGEMENT.

(PART 1.)

TAKING CHARGE.

1. The first duty of an engineer in taking charge of a power plant is to make a thorough inspection of all parts of it in order to become familiar with every detail of the engines, boilers, pumps, and their appurtenances. He should notice particularly if any parts of the engines, pumps, etc., such as cylinder heads, valve-chest covers, pump or condenser bonnets, connecting or eccentric rods, are disconnected or removed. If any parts have been removed, advantage should be taken of the opportunity to examine them. The condition of the interior parts of the engines should be noted. If the cylinder heads are off, he should examine and try with a wrench the follower bolts and piston-rod nuts. He should look carefully at the walls of the cylinders, and if they are cut, grooved, or pitted, he should make a note of the fact for future reference and comparison. The clearance between cylinder head and piston should be measured and marked on the guides with a center punch in order to ascertain and preserve a record of the space there is to spare for taking up the wear on connecting-rod journals. The cylinders should be wiped out thoroughly with oily waste and a liberal coating of cylinder oil should be applied to the wearing surfaces before closing up the cylinders.

2. If the valve-chest covers are off, the condition of the valves and seats should be noted. The lead of the valves should be examined; this may be done by turning the engines over by hand, having first poured a little oil into the journals. The valve gear should be watched during the operation of turning the engine, in order to discover if there is any derangement of the valve gear; if there is any obstruction to the engine turning freely, it will be revealed during the turning process.

3. If the pump-valve bonnets are off, the valves and valve chambers should be examined. If the valves are of hard, flat rubber and any of them are much worn on their lower faces, they should be turned over; if they are curled from standing dry for some time, or other causes, they should be faced off.

4. If the condenser bonnets are off, examine the interior of the condenser in order to ascertain its condition. If a surface condenser, fill the steam side of the condenser with water and look for leaky tubes; replace with new ones any tubes that may be split and renew all leaky tube packing.

5. Just before replacing any of the covers, bonnets, or cylinder heads, a final examination should be made to see that no tools, waste, or other foreign matter are left inside; look particularly for monkeywrenches, hammers, cold chisels, hand lamps, and pieces of waste, as it is quite a usual occurrence to find one or more of these articles inside of the machinery, left there by careless workmen; it is advisable that the engineer in charge should attend to this duty personally. Prior to replacing covers, bonnets, or cylinder heads, the gaskets should be examined; if they are torn or worn out, new ones should be used, and a thin coating of black lead (graphite) should be applied to them before they are put in place.

6. If the connecting-rod is disconnected at the crankpin end, an excellent opportunity to examine the condition of

the crankpin and brasses presents itself; if they are cut or are rough, they should be scraped down with a scraper and finished off with a smooth file, but it must be skilfully done; in connecting up, the crankpin brasses should not be set up too tight upon the pin; better leave them a little slack, to be taken up after the engine has been running for awhile.

7. Examine the piston rod and valve-stem stuffingboxes and put a turn or more of packing in them, as may be required. Examine the cylinder relief valves, if any are fitted to the engine; also examine the drain cocks to see that they are not stopped up.

8. Having put the engine in good order and gotten it ready for steam, the engineer should turn his attention to the feedwater apparatus; he should examine the main feed-pump most carefully. If it is a plunger pump, note the condition of the plunger packing and repack the stuffing-box if necessary; if it is a piston pump, examine the piston-rod packing and put in a turn or two of packing if the box will take it. Examine the petcock and see that it is not stopped up.

9. When satisfied that the pump is in good order, proceed to trace up the pipes. Commence with the suction pipe at the pump; follow it up and examine every foot of it to and from the filter, feedwater heater, or grease extractor, if there be one or the other, to the source of the feedwater supply, wherever that may be, and make sure that there is nothing the matter with this supply. Now start at the pump again and follow up the delivery pipe, tracing it through all its windings, noting every bend and connection, if any; also note where it enters and leaves the feedwater heater, purifier, or economizer, as the case may be, then on through the check-valve and globe valve to the point where it enters the boiler; take out the check-valve and carefully examine it, as well as its seat; wipe off the valve and valve seat with oily waste, and if it is in good condition, replace

the valve, but if found in poor order, repair it or replace it with a new one immediately. Try the globe valve in the feedpipe; if it works stiffly, oil the stem and thread and run the valve up and down until it works freely. Treat all globe valves in a similar manner, and if any of them need packing, attend to it at once. Trace up the auxiliary feedpipes, both suction and delivery, in the same manner and with the same care and attention that was given the main feedpipes.

10. Trace up, from beginning to end, all the auxiliary steam and exhaust piping to and from the various auxiliary engines and pumps, neglecting nothing. Note if the exhaust pipes of the auxiliary system lead to a condenser; if so, locate the valves for changing the auxiliary machinery from non-condensing to condensing and vice versa.

11. If the main engines are condensing, examine the air pump and circulating pump and their valves and trace up all steam pipes and water pipes leading to and from them, whether the pumps are operated by the main engine or independently.

12. Locate all cocks and valves in the piping and ascertain what every one of them is for; locate particularly all those connected with the feedwater supply system, both in the steam pipes and in the water pipes.

13. If the plant is a modern one and is supplied with the latest economical and other appliances and apparatus, such as a separator, grease extractor or filter, feedwater purifier, heater or economizer, evaporator, superheater, reheaters, etc., they must be included in the preliminary inspection and should receive the same care and attention as the other parts of the machinery.

14. In some plants the water of condensation from the steam pipes, valve chest, cylinders, steam jacket, and wherever else it may collect, is saved for use in the boilers;

this is accomplished by leading the various drain pipes into a manifold, from which a single pipe conveys the water to a steam trap, and from thence, by another pipe, to the hot-well or feed tank, where it mingles with the feedwater. These drain pipes should be traced up and examined from beginning to end. Look for leaky or broken joints and split pipes; if any are found, they must be repaired at once. Look also for badly rusted places in the pipes. Note if they are exposed to unusual dampness or dripping water; if such is the case, a coat of thick red-lead paint or paraffin varnish will afford considerable protection. Similar treatment should be accorded to any other iron or steel piping under floors or in places not easily accessible.

15. If during this inspection anything is found that is out of order, it should be repaired at once. Tools and stores for the ordinary repairs are generally provided.

STARTING AND STOPPING ENGINES.

INTRODUCTION.

16. Owing to the great variety of engines and valve gears in use and to the great difference in the sizes and power of steam plants, involving a wide range of appliances and apparatus, it is not possible to give specific directions in detail for starting and stopping each and every one of them. General instructions, with a few examples, can only be given, but it is the intention to make them full enough to enable the intelligent engineer, by using a little judgment and discretion, to apply them to all types of engines.

GENERAL INSTRUCTIONS.

17. Warming Up and Getting Ready.—The engine having previously been put in thorough order and the fires having been lighted in the boiler, it is assumed that the

steam pressure is now approaching its working point. About 15 or 20 minutes before starting the engine, raise the stop valves just off their seats and let a little steam flow into the steam pipe; open the drain cock on the steam pipe just above the throttle. When the steam pipe is thoroughly warmed up and steam blows through the drain pipe, close the drain cock and open the throttle just enough to let a little steam flow into the valve chest and cylinder, or use the by-pass around the throttle, if one is fitted. Open the cylinder relief valves (or drain cocks), also the drain cocks on the valve chest and exhaust pipe, if a non-condensing engine. If cylinders are jacketed, turn the steam into the jackets and open the jacket drain cocks. While the engine is warming up, fill the oil cups and sight-feed lubricator. Squirt a little oil into all the small joints and journals that are not fitted with oil cups. Wipe off the guides with oily waste and squirt some oil over them. By this time the engine is getting warm; if fitted with by-pass valves, use them to admit steam into both ends of the cylinder. Further operations of warming up the cylinder will depend somewhat on the type of engine and valve gear; therefore, additional instructions regarding this matter will be given under their respective headings. In general, however, all cylinders, especially if they are large and intricate castings, should be warmed up slowly, as sudden and violent heating of a cylinder of this character is very liable to crack the casting by unequal expansion.

18. An excellent and economical plan for warming up the steam pipe and the engine is to open the stop-valves and throttle valve at the time or soon after the fires are lighted in the boilers, permitting the heated air from the boilers to circulate through the engine, thus warming it up gradually and avoiding the accumulation of a large quantity of water of condensation in the steam pipe and cylinder. When pressure shows on the boiler gauge or steam at the drain pipes of the engine, the stop-valves and throttle may be closed temporarily, but not hard down on their seats. When this

method of warming up the engine is adopted, the safety valves should not be opened while steam is being gotten up.

19. In attending to these preliminary arrangements certain precautions should be taken. For example, stop-valves and throttle valves should never be opened quickly or suddenly and thus permit a large volume of steam to flow into a cold steam pipe or cylinder. If this is done, the first steam that enters will be condensed and a partial vacuum will be formed. This will be closely followed by another rush of steam with similar results, and so on until a mass of water will collect, which will rush through the steam pipe and strike the first obstruction, generally the bend in the steam pipe near the cylinder, with the force of a steam hammer, and in all probabilities will carry it away and cause a disaster. This is called a **water hammer** and has caused many serious accidents.

20. Another precaution that should be taken is the easing of the throttle valve on its seat before steam is let into the main steam pipe; otherwise, the unequal expansion of the valve casing may cause the valve to stick fast and thereby give much trouble. Even if a by-pass pipe is fitted around the throttle, it would be better not to depend on it. Considerable space has been devoted to the subject of warming up and draining the water out of the steam pipe and engine on account of its importance. Water being non-compressible, it would be an easy matter to blow off a cylinder head or break a piston if the engine were started when there was a quantity of water in the cylinder.

21. The last thing for the engineer to do before taking his place at the throttle preparatory to starting the engine, provided he has no oiler, is to start the oil and grease cups feeding. It is well to feed the oil liberally at first, but not to the extent of wasting it; finer adjustment of the oiling gear can be made after the engine has been running a short time and the journals are well lubricated.

22. Cleaning Up.—After an engine has been stopped after a run and everything has been made secure, the machinery should be wiped off before the oil has had time to set. Both the bright work and the painted parts should receive attention in this respect. If there are any rusty spots on the bright work, they should be immediately scoured off with emery cloth. The floors should also be cleaned up and all dirt gathered together and consigned to the ash heap. Cleanliness is essential to a well-kept engine room, and the grade and value of the engineer in charge of it can very readily be determined by a glance of the practiced eye around the engine room. Oil should not be spilled or spattered about; there is no necessity for it and it is a waste of oil. Drip pans should be placed wherever they will do any good and they should be emptied and cleaned out at least once a day. Much saving in oil bills will be effected by the use of an efficient oil filter to filter the drip oil and use it over again instead of throwing it away.

23. When flour emery or emery cloth is used for cleaning bright work, the greatest care should be exercised not to let any of the grit get into the journals; they will be sure to cut if any substance of a gritty nature gets into them. All the oil holes in the small joints and journals that are not fitted with oil cups should be plugged up immediately after the engine is stopped and kept closed until the engine is ready to be started again. Emery should not be used to polish brass or composition; Bath brick is much better for this purpose.

STARTING A SLIDE-VALVE NON-CONDENSING ENGINE.

24. Assuming that the general instructions given in Arts. **17** to **23** have been complied with, the engineer should now take his place at the throttle, having first opened wide the stop-valves. The drain cocks on the steam pipe and engine are supposed to be open and the throttle valve just off its seat. Some steam has been allowed to enter the

valve chest and the cylinder is partly warmed up; it is now the duty of the engineer to ascertain if steam has entered both ends of the cylinder and that both ends of it are heated equally. As both steam ports cannot be open to the steam at the same time, the engine, if not provided with a by-pass, should be turned by hand so that both steam ports are opened alternately, thus admitting steam to both ends of the cylinder. In turning the engine, finally stop when the crank is on its upper or lower half center, that being the best point from which to start the engine. When it is evident that all condensation of steam has ceased in the steam pipe, valve chest, and cylinder and all the water has been blown out of them, the engine is ready to be started.

25. A slide-valve non-condensing engine is started by simply opening the throttle; this should be done quickly in order to jump the crank over the first center, after which the momentum of the flywheel will carry it over the other centers. The engine should be run slowly at first, gradually increasing the revolutions to the normal speed. When the engine has reached full speed, the drain pipes should be examined; if dry steam is blowing through them, close the drain cocks; if water is being delivered, let the drain cocks remain open until steam blows through and then close them.

26. The engineer should now make a trip around the engine to ascertain if the journals are running cool. First, try the crankpin end of the connecting-rod by touching it with the palm of the hand; to do this safely, on a high-speed engine, requires some skill and experience, but the art can be acquired by a little practice; the beginner, however, should be very cautious that he does not get his hand caught in the machinery. If the end of the connecting-rod is only blood warm, no harm has yet been done, but it is an intimation that the crankpin may get hot, and requires watching. Assuming that the crankpin is running cool, the next step is to feel the shaft journals and examine the lubricating apparatus. If the journals are running cool, decrease the

oil feed gradually and carefully until there is just enough oil fed into the journals to supply the demand without unnecessary waste. It is supposed that the engine is now running satisfactorily, and the engineer may hence turn his attention to a general inspection of his department.

27. It sometimes happens that a cracking noise is heard in the cylinder after the engine has been running for a while. This means "water in the cylinder," and the cylinder drain cocks should be opened promptly. It is also an intimation that the boiler is inclined to prime; this may be checked by closing the main stop-valve just enough to wiredraw the steam a little.

STOPPING A SLIDE-VALVE NON-CONDENSING ENGINE.

28. To stop a slide-valve non-condensing engine, it is only necessary to shut off the supply of steam by closing the throttle, but care should be taken not to let the engine stop on the center. After a little practice, the beginner will be able to stop the engine at any desired point of the revolution. No rule can be laid down for this; it is entirely a matter of experience.

29. After the engine is stopped, shut off the oil feed and close the main stop-valve; be sure that the valve is seated, but without being jammed hard down on its seat. The drain cocks on the steam pipe and engine may or may not be opened, according to circumstances. It will do no harm to allow the steam to condense inside the engine, as the engine will then cool down more gradually, which lessens the danger of cracking the cylinder casting by unequal contraction. All the water of condensation should be drained from the engine before steam is again admitted to it.

30. When an engine is required to run in either direction, in answer to signals or otherwise, as in the case of

hoisting engines and locomotives, it is usually fitted with the link-valve motion, which is operated by a system of levers or other apparatus called the **reversing gear.** In warming up an engine fitted with a link, it is only necessary to run the link up and down if a horizontal engine, or back and forth if a vertical engine, to admit steam into both ends of the cylinder; and to start or stop such an engine, either the go-ahead or the backing eccentric, as required, is thrown into action by operating the link by means of the reversing gear.

STARTING A SLIDE-VALVE CONDENSING ENGINE.

31. A steam engine of the condensing type is fitted with either a surface, a jet, or an injector condenser. The function of a condenser is to convert the exhaust steam from the engine into water by condensation, thereby producing a vacuum in the condenser. The pressure of the atmosphere is thus partially removed from the exhaust side of the piston and the net pressure correspondingly increased.

32. If the engine is fitted with a surface condenser, the condenser will be supplied with an air pump and a circulating pump. The air pump removes the air, vapor, and water of condensation from the condenser; it discharges the water into the hotwell or feed tank, while the air and vapor escape into the atmosphere. The circulating pump supplies the condensing water and forces it through the tubes of the condenser.

33. It is sometimes the case that the air pump and the circulating pump are attached to and operated by the main engine; more frequently, however, they are operated by a separate and independent steam cylinder, in which cases the apparatus as a whole, including the condenser, is called the **vacuum engine.**

Another arrangement of these pumps is the following: A centrifugal circulating pump is used instead of a reciprocating pump, by which the circulating pump can be operated independently of the air pump, permitting the speed of the circulating pump to be changed without affecting the speed of the air pump. This is desirable, because a greater quantity of injection or condensing water is required in summer than in winter on account of its higher temperature at that season of the year.

34. Before the main engine is started, the air pump and circulating pump should be put into operation and a vacuum formed in the condenser; this will materially assist the main engine in starting promptly, and in cases where the engine is worked to bell signals, such as a hoisting engine in a mine or elsewhere, this is a most important consideration. Prior to starting the air and circulating pumps, the injection valve should be opened to admit the condensing water into the circulating pump; the delivery valve should also be opened at this time. The same course of procedure that is used in warming up and draining the water out of the main engine should be followed with the vacuum engine, and it is started in the same manner, i. e., by simply opening the throttle.

35. After the main engine has been running for a few minutes to equalize temperatures, the speed of the air and circulating pumps and the admission of injection water should be regulated so as to maintain about 26 inches of vacuum in a surface condenser and a feedwater temperature of about 115° F. A higher vacuum than 26 inches, when the barometer stands at 30 inches, will result in a loss of heat from cold feedwater, and it will also cause a high-speed engine to thump while passing the centers through insufficient compression or cushion for the piston; a lower vacuum than 26 inches will cause a loss by too much back pressure. As a rule, there should be about 2 pounds (absolute) of back pressure on the exhaust side of the piston;

this is equivalent to 4 inches on the vacuum gauge and a feedwater temperature of about 115° F.; therefore, the reading of the vacuum gauge should be about 4 inches below the reading of the barometer to get the best results from the engine.

36. If an ordinary jet condenser is used, no circulating pump is required, the water being forced into the condenser by the pressure of the atmosphere. If the air pump is operated by the main engine, which is sometimes the case, a vacuum will not be formed in the condenser until after the engine is started and at least one upward stroke of the air pump is made. In this case the injection valve must be opened at the same moment the engine is started; otherwise the condenser will get "hot" and a mixture of air and steam will accumulate in it and prevent the injection water from entering. When this occurs it is necessary to pump cold water into the condenser by one of the auxiliary pumps through a pipe usually fitted for that purpose; if such a pipe has not been provided, it may be found necessary to cool the condenser by playing cold water upon it through a hose.

37. Jet condensers are sometimes fitted with a valve that automatically opens outwards, called a **snifting valve,** or **snifter,** by which the accumulated steam, vapor, and air may be discharged into the atmosphere. This valve is a disk of metal, similar to a safety valve, that is held to its seat by its own weight and the pressure of the atmosphere. It serves to relieve the condenser of pressure in case of an accident to the air pump.

38. If the engine is fitted with a jet condenser, the course of procedure in starting is similar to that followed in starting an engine with a surface condenser, viz.: The air pump should be started before the main engine is started and thereby form a vacuum in the condenser beforehand, and it should be stopped after the main engine is stopped.

STOPPING A SLIDE-VALVE CONDENSING ENGINE.

39. The operation of stopping a slide-valve surface-condensing engine is precisely similar to that of stopping a non-condensing engine of the same type, with the addition that after the main engine is stopped the vacuum engine is also stopped, and in the same way, i. e., by closing the throttle, after which the injection valve and the discharge valve should be closed and the drain cocks opened.

40. With a jet condenser, the operation of stopping the engine is the same as the above, with the exception that the injection valve should be closed at the same moment that the engine is stopped.

SUMMARY.

41. The instructions given in Arts. **24** to **40** apply to any slide-valve engine, whether vertical or horizontal, and also whether it is fitted with a ball or pendulum governor, a shaft automatic governor, or if it is without any governor at all, from the fact that a governor acts only when the engine is running at or near its normal speed; therefore, while starting or stopping an engine the governor is not in action.

STARTING A SIMPLE CORLISS ENGINE.

42. A simple engine fitted with Corliss valves and valve gear is started and stopped in a somewhat different manner from that practiced with the plain slide-valve engine.

43. In the Corliss engine the eccentric rod is so constructed and arranged that it may be hooked on or unhooked from the eccentric pin on the wristplate at the will of the engineer. The wristplate is provided with a socket for the starting bar; the starting bar may be shipped or unshipped as required.

44. In starting an engine of the Corliss type, after all the preliminaries, such as warming up and draining the water from the engine, starting the oil feed, etc., as heretofore explained, have been attended to, the starting bar is shipped into its socket in the wristplate and the throttle is opened. The starting bar is then vibrated back and forth by hand, by which the steam and exhaust valves are operated through the wristplate and valve rods; as soon as the cylinder takes steam the engine will start. After working the starting bar until the engine has made several revolutions and the flywheel has acquired sufficient momentum to carry the crank over the first center, let the hook of the eccentric rod drop upon the pin on the wristplate. As soon as the hook engages with the pin, unship the starting bar and place it into its socket in the floor.

45. The way to determine in which direction the starting bar should be first moved to start the engine ahead is to note the position of the crank, from which the direction in which the piston is to move may be learned. This will indicate which steam valve to open first; it will then be an easy matter to determine in which direction the starting bar should be moved. After a little practice the engineer will know at a glance which way to work the starting bar.

46. The engine having been started, the engineer should attend to those duties that have been mentioned in the instruction for the slide-valve engine under similar circumstances.

47. If the engine is of the condensing type, the same course of procedure in starting the vacuum engine should be followed as with the simple slide-valve condensing engine, which has been previously explained. In warming up the cylinder of a Corliss engine, it is not necessary to turn the engine to admit steam to both ends; it is only necessary to work the valves by hand with the starting bar.

STOPPING A SIMPLE CORLISS ENGINE.

48. A Corliss engine is stopped by closing the throttle and unhooking the eccentric rod from the pin on the wristplate; this is done by means of the unhooking gear provided for the purpose. As soon as the eccentric rod is unhooked from the pin, ship the starting bar into its socket in the wristplate and work the engine by hand to any point in the revolution of the crank at which it is desired to stop the engine. Then proceed as directed for the simple slide-valve engine. After stopping a Corliss condensing engine, the same course should be followed as with a slide-valve condensing engine in regard to draining cylinders, closing stop-valves, etc.

STARTING A COMPOUND ENGINE.

49. Before starting a compound engine, the high-pressure cylinder is warmed up in the same manner as a simple engine. To get the steam into the low-pressure cylinder is an operation, however, that will depend on circumstances. If the cylinders are provided with pass-over valves, it will only be necessary to open them to admit steam into the receiver and from thence into the low-pressure cylinder. If the cylinders are not fitted with pass-over valves, the steam can usually be worked into the receiver and low-pressure cylinder by operating the high-pressure valves by hand. Sometimes compound engines are fitted with starting valves, which greatly facilitate the operations of warming up and starting. Usually a compound engine will start upon opening the throttle.

50. It sometimes happens that the engine will refuse to start from various causes, viz.: The high-pressure crank may be on either dead center; there may be too low a steam pressure in the receiver; the engine may be stiff from standing idle for a long time, and the oil in the journals has become gummy; the pistons may be rusted fast in the cylinders, or the cylinders may not have been wiped out

after the last run and a coating of carbonized oil and rust may have collected on their walls and caused the pistons to stick fast. If the engine is fitted with independent adjustable cut-offs, the cut-offs may be set to cut off too early; or there may be water in the cylinders. There may be some obstruction to the engine turning, although that matter is supposed to have been attended to during the preliminary inspection. In most cases the conditions will suggest the remedy. If the high-pressure crank of a cross-compound engine is on its center and the low-pressure engine will not pull it off, it must be jacked off. Ordinarily, it will be found that upon admitting steam of sufficiently high pressure into the receiver, the low-pressure piston will move and take the high-pressure crank from off the center; if the pressure in the receiver is too low to start the low-pressure piston, more steam must be admitted into the receiver. If the engine is stuck fast from gummy oil or rusty cylinders, all wearing surfaces must be well oiled and the engine jacked over at least one entire revolution. If the cut-offs are run up, run them down, full open. If there is water in the cylinders, blow it out through the cylinder relief or drain valves, and if there is any obstruction to the engine turning, remove it.

51. If the crank of a tandem compound engine is on the center, it must be pulled or jacked off. If the high-pressure crank of a cross-compound engine is on the center, it may or may not be possible to start the engine by the aid of the low-pressure cylinder, depending on the valve gear and crank arrangement. When the cranks are 180° apart (which is a very rare arrangement), the crank must be pulled or jacked off the center. When the cranks are 90° apart and a pass-over valve is fitted, live steam may be admitted into the receiver and thence into the low-pressure cylinder, in order to start the engine. When no pass-over is fitted, but the engine has a link motion, sufficient steam to pull the high-pressure crank off the center can generally be worked into the low-pressure cylinder by working the

links back and forth. When no pass-over is fitted, but the high-pressure engine can have its valve or valves worked by hand, steam can be gotten into the low-pressure engine by working the high-pressure valve or valves back and forth by hand. If no way exists of getting steam into the low-pressure cylinder while the high-pressure crank is on a dead center, it must be pulled or jacked off.

52. If the air and circulating pumps are attached to and operated by the main engine, a vacuum cannot be generated in the condenser until after the main engine has been started. Consequently, in this case, there is no vacuum to help start the engine; therefore, if it is tardy or refuses to start, it will be necessary to resort to the jacking gear and jack the engine into a position from which it will start. With an independent vacuum engine, however, it is seldom that any such difficulties in starting an engine are encountered. A vacuum having been generated in the condenser beforehand, the pressure in the receiver acting on the low-pressure piston causes the engine to start very promptly, even though the high-pressure crank may be on its center. Notwithstanding differences of opinions among designers in regard to this matter, the value of an independent vacuum engine is fully appreciated by the man at the throttle. In fact, it is almost indispensable with compound condensing engines of high power that are required to run in either direction in answer to bell or other signals, such as hoisting, rolling-mill, and marine engines, where promptness in starting is absolutely essential.

53. Large reversible engines are usually fitted with steam starting and reversing gears, each builder suiting his own fancy in regard to the design; therefore, the engineer should promptly familiarize himself with the mechanism of the particular starting gear that he has to handle.

54. If the engine is fitted with adjustable cut-offs, they should be so manipulated, after the engine has been started,

that it will run smoothly and at the lowest steam consumption attainable with the given load. The only way of finding the proper points of cut-off is by experiment, setting the cut-offs where judgment and experience dictate and noting the effect upon the smooth running and steam consumption.

STOPPING AND REVERSING A COMPOUND SLIDE-VALVE ENGINE.

55. Compound slide-valve engines, whether condensing or non-condensing, are stopped by closing the throttle, and, if a reversing engine, throwing the valve gear into mid-position. If the stop is a permanent one, the usual practice of draining the engine, steam chests, and receiver, closing stop-valves, stopping the oil feed, etc. may be followed; or, as before stated, if the cylinders and receivers are complicated castings, as they are apt to be in an engine of this kind, it would be better not to drain them while they are hot, but to let them cool down gradually to avoid the danger of cracking the castings from too sudden and, therefore, unequal contraction.

56. If the engine is intended to run in both directions in answer to signals, as in the cases of hoisting, rolling-mill, and marine engines, the operator, after stopping the engine to signal, should immediately open the throttle very slightly, in order to keep the engine warm, and stand by for the next signal. If the engine is fitted with an independent or adjustable cut-off gear, it should be thrown off, i. e., set for the greatest cut-off, for the reason that the engine may have stopped in a position in which the cut-off valves in their early cut-off positions would permit little or no steam to enter the cylinders, in which case the engine will not start promptly, and perhaps not at all. While waiting for the signal, the cylinder drain valves should be opened and any water that may be in the cylinders blown out. When dry steam blows through the drains, the cylinders are clear of water.

57. When the signal to start the engine is received, it is only necessary to throw the valve gear into the go-ahead or backing position, as the signal requires, and to operate the throttle according to the necessities of the case, for which no rule can be laid down beforehand, as the position of the throttle will depend on the load on the engine at the time. Handling the throttle must be learned by experience on the spot.

58. It is frequently the case that in large plants a working platform is provided on which the reversing gear, throttle-valve lever or wheel, cylinder drain-valve levers, and all other hand gear, gauges, etc., are located and placed within easy reach of the engineer's station. This platform is usually placed in a commanding position, from whence the engineer has a full view of the moving parts of the engine. This is a matter of considerable importance, although an experienced engineer, after he becomes familiar with the various sounds produced by his engine under different conditions, will depend on the ear as much as on the eye in running it. In most cases any derangement of the machinery will give timely warning by making an unusual sound; perhaps it may be only a slight clicking noise, scarcely noticeable among so many different sounds. A careful engineer, however, can detect it as quickly as an expert musician can detect a discordant note, and he should at once proceed to find out the cause, thereby anticipating and preventing a possible breakdown.

STARTING AND STOPPING A CORLISS COMPOUND ENGINE.

59. The operation of starting and stopping a Corliss compound engine is precisely similar to that of starting and stopping a Corliss simple engine; the high-pressure valve gear only is worked by hand in starting, the low-pressure eccentric hook having been hooked on previously. The low-pressure valve gear is only worked by hand while

warming up the low-pressure cylinder. The same directions that were given for operating the simple condensing engine apply to the Corliss condensing engine, so far as the treatment of the air pump, circulating pump, and condenser is concerned.

STARTING, STOPPING, AND REVERSING TRIPLE- AND QUADRUPLE-EXPANSION ENGINES.

60. The management of triple- and quadruple-expansion engines is the same as that practiced with the compound engine, with the exception that there is a greater number of moving parts, more journals, more hand gear, and more machinery, in general, to look after, requiring greater activity and alertness on the part of the engineer to care for it.

LINING ENGINES.

INTRODUCTION.

61. Purpose.—The operation of lining an engine is for the purpose of locating the different parts in relation to each other, so that no undue strains will be exerted on any one of the parts and the friction of the moving parts will be reduced to a minimum. Absolute accuracy, though desirable, cannot be attained, and if it could be attained, it could not be maintained. Too much care, however, cannot be exercised in lining an engine, as its future smooth running and efficiency will depend very largely on the accuracy with which this operation is performed.

62. Requirements.—An engine, in order to be in line, must fulfil the following requirements:

1. The center line of the shaft must be at right angles to the center line of the cylinder.

2. The wearing surfaces of the guides must be parallel to the center line of the cylinder. When two guides are used, they must be parallel to each other, and, at least in most designs, equidistant from the center line of the cylinder.

3. The center line of the wristpin must be at right angles to the center line of the cylinder and must lie in the same plane.

4. The center line of the crankpin must be parallel to the center line of the shaft.

5. The center line of the cylinder and of the shaft must both lie in the same plane.

6. The center line of the bore of the brasses at both ends of the connecting-rod must be parallel to one another and must be at right angles to the center line of the connecting-rod.

7. The center line of the piston rod must coincide with the center line of the cylinder.

If the above requirements are fulfilled, the engine may be said to be in line, as far as the machine itself is concerned. In addition to the requirements enumerated above, it is generally necessary that the crank-shaft be level.

LINING UP.

63. Preliminary Conditions.—Let it be understood that a new single-cylinder, horizontal engine is in the course of erection or that an old engine of the same type is receiving a thorough overhauling. In the case of the new engine, it is assumed that the foundation has been built, the bedplate placed in its proper position and secured there by the anchor bolts, and the cylinder has been secured to the bedplate. As the cylinder was fitted to the bedplate in the shop, it may be assumed that it is correctly placed. In the case of an old engine being overhauled, it is understood that all the moving parts have been removed and that the cylinder heads are off. The condition of both engines are now supposed to

be the same; therefore, henceforth the course of procedure will be the same for both.

64. Stretching Center Line of Cylinder.—The first step in lining an engine is stretching a line coincident with the center line of the cylinder. This may be done in the following manner:

A strip of board or other convenient material is secured to the head end of the cylinder by means of the stud bolts and nuts, as shown at *a*, Fig. 1. A hole about 1 inch in diameter

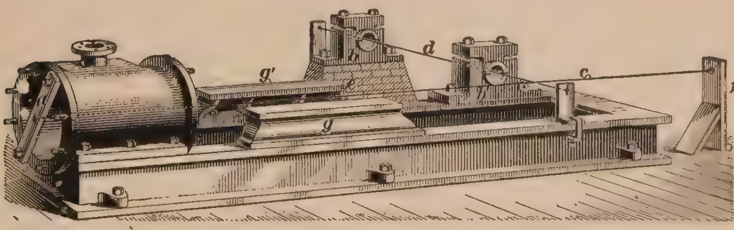


FIG. 1.

is bored through the strip approximately in line with the center of the cylinder. Some form of standard, as *f* for instance, which is pierced similarly to the strip of board *a*, should be erected at the crank end of the bedplate. A very fine braided cord or piece of thin annealed copper wire, as *c*, may now be stretched very tightly through the holes in *a* and *f*. In order to allow of ready adjustment, each end of the line may be fastened to the middle of a piece of, say, ¼-inch round iron about 2 inches long, or, better yet, be passed through a hole in the center of a piece of stout sheet tin or other metal. A knot at each end of the line will prevent it slipping through the holes. The pieces of sheet metal may be cut, say, 2 inches square, and the diagonally opposite corners may be turned up at right angles to form handles by which they may be adjusted. If the line used is a fine wire, two holes may be punched in each piece of sheet metal and the end of the wire passed through one of the holes, brought back through the other in the form

of a loop, and the end stopped off around the main part of the line.

65. The line should be set central to the bore of the cylinder at the head end by calipering from the inside of the cylinder counterbore to the line. This may be done with a pair of inside calipers, but in most cases it is better to use a light pine stick, like that shown in Fig. 2. The stick *a* should be about 1 inch shorter than the radius of the cylinder counterbore and tapered at each end, with a thin 1-inch

FIG. 2.

wire nail driven into each end, as shown at *b*, making the total length of the stick, including the nails, the exact radius of the counterbore. The advantage of the stick in calipering is that it is lighter and more convenient to use than inside calipers.

If the calipers or stick will just touch the line, no matter from which point on the circumference of the counterbore the measurement is taken, the head-end part of the line will be central to the bore of the cylinder, provided the measurements were carefully and accurately made. If the measurements do not agree, the line that passes through *a*, Fig. 1, must be shifted in the direction shown by the variation until it coincides with the center line of the cylinder. It is considered good practice to make four measurements 90° apart.

66. After adjusting the line at the head end of the cylinder, the crank-end part of the line may be trued up in a similar manner by moving the line at the standard *f*, Fig. 1. The line, now, may or may not be properly adjusted. Hence, to make sure, the alinement of the line at the head end should be tried again. Now, unless the line happened to be very close to the center line of the cylinder before any adjustments were made, it will usually be found to be a shade out of the center, and hence requires readjustment. After adjusting the head-end part of the line, try

the crank-end part again and adjust it. Continue this practice, first at one end of the cylinder and then at the other, until no further adjustment is necessary or possible. Then, if the measurements have been carefully made, the line *c* may be considered to coincide with the center line of the cylinder.

67. If the crank-end head is cast solid with the cylinder, as is frequently the case, the line *c* must be trued up from the bore of the piston-rod stuffingbox. This may be done by means of a stick similar to that shown in Fig. 2, but it may be more conveniently done by means of the device shown in Fig. 3. This device consists of a hardwood block *a*, which is turned to just fit into the stuffingbox and has a $\frac{1}{4}$-inch hole *b* bored in the exact center. The face of the block is faced square with the outside, and two center lines *c d* and *e f* are drawn across the face at right angles to each other. By sighting along these lines, it is easy to determine when the line or wire is central with the stuffingbox.

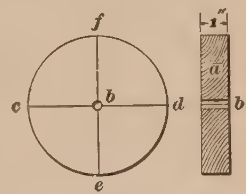


FIG. 3.

68. Stretching Center Line of Shaft.—After completing the adjustment of the line *c*, Fig. 1, another line, as *d*, is stretched to coincide with and to represent the center line of the shaft; it *must* be at right angles to the line *c* and generally *must* be level.

In many engines, the outboard bearing *b*, Fig. 1, is movable to a certain extent and may be adjusted in regard to its relative position with the inboard bearing *b'*. The inboard bearing, however, is usually cast solid with the bedplate; therefore, it is fixed and cannot be moved. Hence, in adjusting the line *d*, the aim must be to stretch the line through the center of the bearing *b* and at the same time to have it at right angles to the line *c*. In order to accomplish this, it will be necessary to erect two standards, as shown in Fig. 1, to which to fasten the line. It is supposed that the top brasses and caps of the main-shaft bearings are in place

just as they would be if the shaft itself were in place. The alinement of the line *d* in reference to the bearing *b'* can be tested by calipers, but a better method is to insert a block of wood, made as shown in Fig. 3, into the bore of the bearing. It would be well to fit a similar piece of wood into the bore of the outboard bearing also, as it will greatly facilitate the adjusting of the line *d* by passing it through the holes in the centers of these blocks.

69. Squaring Center Line of Shaft.—We may now proceed to test the angle between the two lines *c* and *d*, Fig. 1. The line *d* may be approximately squared with the line *c* by a carpenter's square pressed gently against the two lines. Great care in the use of the square is necessary, since the lines will yield to quite an appreciable extent under a very slight pressure. A crowding of some part of the square against either line will deflect it and seriously interfere with the test. If the square shows that the lines are not at right angles to each other, the line *d* should be shifted until they are, always keeping in mind the fact that the line must coincide with the center line of the inboard bearing.

70. In lining engines of the larger sizes, the carpenter's square is not accurate enough, since its legs are very short in proportion to the length of the lines. A somewhat different method may then be used, which is based upon the principles of geometry.

Procure a slender strip of wood—6, 8, or 10 feet long, according to the extent of the space to work in. Taper the strip to a point, like a lead pencil, at each end; divide the strip into two exactly equal parts and mark it plainly in the middle. Now hold the strip gently along the line *c*, Fig. 1, with the mark in the middle at the line *d*; mark the line *c* at each end of the strip; then put one end of the strip at one of the marks just made on line *c* and the other end of the strip against the line *d* and mark the place where the end of the strip touches the line *d*; repeat this

operation on the other side of the line *c*, and if the end of the strip touches the line *d* at exactly the same spot that it did before, the line *d* is at right angles to the line *c*; but if the end of the strip does not touch the same spot on the line *d* at both trials, the line *d* is not at right angles to the line *c*, and the line *d* must be shifted in the right direction and the whole operation must be repeated until the end of the strip *does* touch the line *d* at the same point at both trials. In shifting the line *d* during these operations, care must be taken that its position through the center line of the inboard bearing *b'* is maintained. A convenient method of marking the lines is to tie a piece of bright-colored thread around the lines at the points that are desired to be marked. The colored threads are easily sighted, and the marks are more sharply defined and cleaner than when made with paint or chalk and they have the advantage of being easily shifted along the lines at will.

71. In order to prevent any displacement of the line while measuring, it is good practice to place blocks of wood or any other convenient material against it at the points marked, in order to steady it, but care must be used not to deflect the line; when the blocks are properly placed, they may be temporarily secured in position.

72. Another test to determine whether the lines are at right angles to each other is to measure from their intersection distances of 6 and 8 feet, one on each line. Then, if the measurement from line to line, measuring in a straight line from the points just laid off, is 10 feet exactly, the lines are at right angles. Instead of using the values 6, 8, and 10 feet, any convenient multiple may be used.

73. Leveling Center Line of Shaft.—The line *d* representing the center line of the crank-shaft may be tested for being level by means of a spirit level, taking great care in applying it not to deflect the line. Another method is to drop a plumb-line from overhead touching the line *d*.

Then, if the line *d* is level, it evidently must be at right angles to the plumb-line, and, consequently, this condition can be tested in the same manner in which the position of line *d* in reference to line *c* was tested.

74. In leveling the line *d*, Fig. 1, it should not be allowed to touch the line *c*, lest it should be deflected; it should pass just over or just under it, say at a distance of $\frac{1}{16}$ inch. If the lines touch each other, any vertical movement of either end of the line *d* will, when in the wrong direction, deflect both lines, thus defeating the primary object of stretching them; viz., that they shall represent the center lines of the cylinder and shaft. After leveling the line *d*, it is well to verify the relative alinement of both lines. When the line *d* is properly adjusted, the fifth requirement will be complied with a degree of accuracy sufficient for practical purposes, providing the lines *d* and *c* will just clear each other where they cross.

75. In lining up a new engine or an old engine with new shaft-bearing brasses, it is generally desirable to have the center line of the shaft lay about $\frac{1}{32}$ inch above the center line of the cylinder, so that as soon as the brasses and journals have worn down to their bearings, the center line of the shaft will be very nearly level with the center line of the cylinder.

76. Shifting Outboard Bearing.—Having proved the correct alinement of the lines *c* and *d*, the outboard bearing may now be shifted until the line *d* coincides with its center line, when it may be secured in position permanently.

77. Testing Alinement of Guides.—We are now ready to test the guides *g*, *g'*, Fig. 1. If the line *d* is level, a spirit level may be used to ascertain if they are in the proper plane in reference to *d*. Their adjustment relative to the center line of the cylinder may be tested in one direction by placing a straightedge successively at each end of the

guides and squarely across them. Then, if the measurements from the lower edge of the straightedge down to the line *c* agree at both ends of the guides, they are in line vertically. If not, the same ends of both guides must either be raised or lowered, remembering that raising one end of the guide is equivalent to lowering the other end. By measuring from the inside edges of the guides to the line *c*, it may be ascertained if the guides are parallel to each other and parallel to the center line of the cylinder. To ascertain if the guides are in a horizontal plane, a spirit level may be placed squarely across the guides at both ends.

78. In lining the guides of a Corliss engine a special device similar to that illustrated in Fig. 4 is often used. This consists of a casting *a* that is turned to fit the inside of

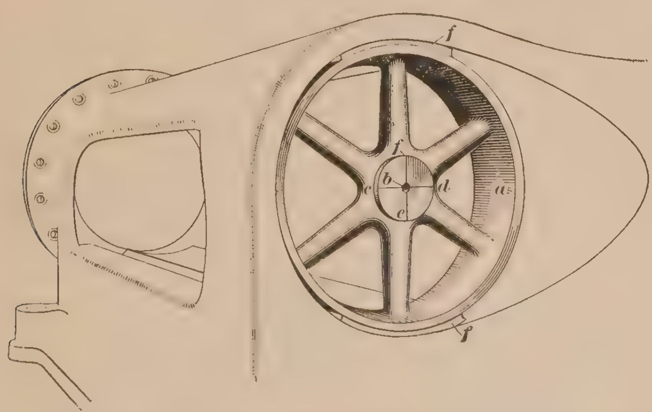


FIG. 4.

the guides. At the center there is a small hole *b* through which the line passes, while the lines *c d* and *e f*, drawn at right angles to each other, serve to locate the center line in its proper position, this being done in a manner similar to that illustrated in Fig. 3 for the piston-rod stuffingbox.

79. Bedding the Shaft.—The line *d*, Fig. 1, having been removed, the crank-shaft, crank, and flywheel are put in place, care being taken not to disturb the line *c*. After the bearings have been adjusted so that the shaft will turn easily in them, the journals of the shaft should be wiped clean and given a coat of red or black marking material. The shaft should then be placed in its bearings, with the lower halves of the brasses in position, and rocked back and forth a few times. The shaft is then lifted out of the bearings and the high spots scraped off with a half-round scraper. This operation is repeated until the shaft shows a good bearing in both the main pillow-block and the outboard bearing. After the lower halves of the boxes are scraped, the upper halves may be put in place and fitted in like manner. The shaft is now lifted from its bearings and the eccentrics and governor-driving device are placed in position, after which the shaft is returned to its place. The crank and flywheel having been fitted to the shaft in the shop, it is to be presumed that they are true with the shaft.

80. Testing Alinement of Shaft.—In order to make sure that the shaft is exactly at right angles to the center line of the cylinder and that it is also level, the following course

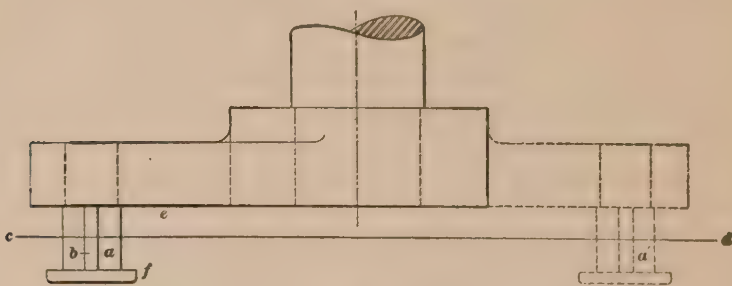


FIG. 5.

may be pursued: The crankpin *a*, Fig. 5, is brought up to the center line *c d* and a piece of wood *b* is fitted between

the face of the crank *e* and the head of the crankpin *f*. A mark is made on this piece of wood to coincide with the line *c d*. The shaft is now given a half revolution to bring the crankpin under the line at the other end of its travel, as shown by the dotted lines at *a'*. If the line on the strip of wood *b* again coincides with the center line *c d*, the shaft is at right angles to the center line of the cylinder.

81. Testing Leveling of Shaft.—In order to test the shaft to see whether or not it is level, a fine plumb-line may be suspended vertically before the shaft at the crank end and the crankpin *a* brought into contact with it at the upper half-center and then tested again at the lower half center. If the end of the crankpin just touches the line at both upper and lower half centers, the shaft is horizontal.

82. Lining the Crosshead.—The line *c*, Fig. 1, having been removed, the piston, piston rod, and crosshead are put in place, centering the piston in the cylinder first of all, when the design of the piston is such as to make this adjustment necessary. The next step to take is to ascertain if the center line of the piston coincides with the center line of the cylinder. This may be done as follows: In Fig. 6, let *d* be the upper surface of the guides, which, when properly alined, lies parallel to the center line of the cylinder; the piston having been previously centered, the center line of the piston rod coincides with the center line of the cylinder; therefore, the piston rod should be parallel with the upper surfaces of the guides. This may be readily tested by placing the piston at the forward end of its stroke; then measure downwards from the lower edge of a straightedge laid across the guides at *e* and *f* to the piston rod. If the

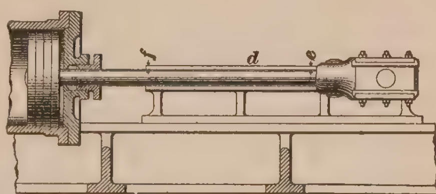


FIG. 6.

two measurements agree, the piston rod is in line; otherwise, the crosshead shoes must be adjusted until the piston rod is in its proper position.

83. Testing the Crankpin.—A method of testing the accuracy of the crankpin is shown in Fig. 7 and it may be put into practice as follows: Connect the connecting-rod to the crankpin and key up the brasses snugly to the pin; then put the crank on or near one of its dead centers, as at c, Fig. 7, and exert a slight pressure against the wristpin end of the connecting-rod in the direction of the arrow x, in order to take up any lost motion in the crankpin brasses. It is understood that the connecting-rod is disconnected from the crosshead, and the latter is pushed back towards the cylinder, so as to be out of the way; then measure the distance a and make a note of it. Put the crank on or near the other dead center c' and measure the distance a'. Then, if the distance a is equal to the distance a', it proves that the center line of the crankpin is parallel to the center line of the shaft in the horizontal plane. But it is necessary that similar conditions should prevail in the vertical plane. To test this, put the crank on the upper half center, as at d, and measure from the wristpin end of the connecting-rod to the inner edge of the guides, as before; then turn the crank to the lower half center d' and measure again from the connecting-rod to the guides. If the measurements agree for both positions of the crank, the crankpin is properly alined. In the figure the crankpin is shown out of line, the connecting-rod then occupying the positions shown in full

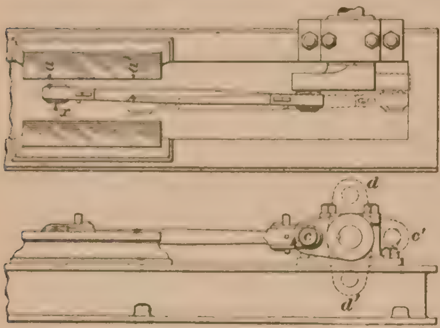


FIG. 7.

and dotted lines, respectively. If the error is very small, it may sometimes be remedied by filing and scraping the crankpin, but if the error is serious, it may require machine work.

84. Testing Bore of Crankpin and Wristpin Brasses. It will generally be advisable to ascertain if the bore of the crankpin brasses is at right angles to the center line of the connecting-rod. This may be done by putting the crank, with the connecting-rod attached to the crank but disconnected from the crosshead, on one of its dead centers, as at *c*, Fig. 7. Then measure the distance from the inner edge of one of the guides to the crosshead end of the connecting-rod, as at *a*, Fig. 7, and make a note of it; then take the rod off the pin and turn the rod half way around on its center line and replace it on the pin. Now measure from *a*, as before; if the two measurements agree, the brasses are correctly bored. To make this test still more satisfactory, the operation may be repeated for several different positions of the crank. The wristpin end of the rod may be tested in like manner; in this case the rod is disconnected from the crankpin and the measurements are taken from the face of the crank or from the collar of the crankpin to the crankpin end of the connecting-rod. If the connecting-rod cannot pass this test satisfactorily, the brasses must be fitted to the crankpin and wristpin by chipping, filing, and scraping until it fills the requirements.

85. Testing Alinement of Wristpin.—The alinement of the wristpin may now be tested, for which purpose the connecting-rod may be used. Key the rod rather snugly to the wristpin, having it disconnected from the crankpin. Then place the crank on or near one of its dead centers and push the crosshead forwards until the end of the connecting-rod just rests on the crankpin. If the center line of the rod is the same distance from both collars of the crankpin, the wristpin is in the correct position in one direction. To

prove it for another direction, put the crank on one of its half centers and repeat the above described operation. If the result is the same as before, the wristpin is in its correct position relative to the center lines of the cylinder and shaft.

86. Testing Alinement of Connecting-Rod Brasses in Reference to Each Other.—It still remains to be proved that the center lines of the crankpin brasses and the wristpin brasses lie in the same plane. This is a very necessary condition, because, otherwise, the brasses when keyed tightly to one pin will not fit the other pin, or, as usually expressed, they will bear on one side only. To make this test, a thin coating of Prussian blue or red-lead paint is put on the crankpin and the rod is connected up and adjusted rather snugly to the wristpin and a little less snugly to the crankpin. The crank is then turned through one revolution, when the crankpin brasses may be examined. If they show marking all over, their correct adjustment is assured, but if the coloring matter is rubbed off at either end of the crankpin, it shows that the brasses do not fit the pin and that they must be filed and scraped until they bear equally on all parts of the pin.

87. Order of Operations.—An engineer in lining up his engine or testing his engine for alinement, will do well to perform the various operations in the same order as given here; he should remember that in order to insure correct results, each part of the engine tested *must* be alined before proceeding further. Thus, it is folly to attempt to prove by the methods given here that the center line of the brasses are in the same plane before the correct relative alinement of the wristpin and crankpin are proven.

If a new engine is so far out of line that it cannot readily be adjusted by liners and scraping brasses while the various parts are being assembled, it should be placed in good order by the builder.

POUNDING OF ENGINES.

CAUSES.

88. The causes of pounding in engines are various; they are not always easy to locate in a large engine, owing to the difficulty of locating the exact source of the sounds. These sounds serve as a warning, however, that something is wrong about the machinery, and no time should be lost in ascertaining where and what it is and taking measures to stop it, thereby preventing a possible breakdown.

LOOSE BRASSES.

89. The most frequent cause of pounding in engines is loose journal brasses; the pounding is produced by the journals striking against the sides of the brasses as the cranks are passing the centers and at the instant the change of direction in the motion of the pistons takes place. If the journal-boxes are very slack, the pounding may be so violent as to cause heating of the journal and boxes by the succession of blows they receive; this may especially occur at the crankpins. The remedy for pounding of this nature is obvious. Stop the engine and set up on the brasses gradually, until, after trial, the pounding ceases, taking great care that they are not set up too tight, else they will heat from friction, which may have a more disastrous effect than a moderate amount of pounding. In the case of shaft journals, they may be set up without stopping the engine, provided they can be reached without danger of the engineer being caught in the machinery.

90. It may so happen that the boxes or brasses are worn down until the edges of the upper half and the edges of the lower half are in contact and cannot be set up on the journal any farther; they are then said to be **brass and**

brass, or brass bound. In a case of this kind, the journal must be **stripped,** as it is called, when the cap and brasses are removed from a journal. The edges of the brasses are then chipped or filed off, in order to allow them to be closed in; the amount to be taken off may be determined by trying the brasses on the journal occasionally or by calipering the journal with outside calipers, transferring the measurement to a pair of inside calipers, with which to measure the bore of the brasses as they are being fitted. It is a most excellent plan in practice to reduce the two halves of the brasses so that they will stand off from each other when in place for a distance of $\frac{1}{8}$ inch to $\frac{3}{16}$ inch and fill this space with hard sheet-brass liners, say from 20 to 22 Birmingham wire gauge in thickness each. The object is this: Should the journal become brass bound, the cap may be slacked off and a pair of the liners slipped out without the necessity of stripping the journal, which it is desirable to avoid whenever possible for the reason that it seems to be impossible in practice to put the journal brasses back just where they were before they were disturbed. In large engines it is almost always the case that journals *will* heat after being stripped, and they require a special watch for several days or until they settle down to their proper position relative to the journal.

91. In some instances journal-boxes are fitted with **keepers,** or **chipping pieces,** as they are sometimes called. These consist usually of a cast-brass liner, anywhere from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch in thickness, having ribs or ridges cast on one side, for convenience of chipping and filing. These keepers are sometimes made of hardwood and are capable of being compressed slightly by the pressure exerted upon them during the setting-up process. When the boxes are babbitted, the body of the box is occasionally made of cast iron, in which case iron liners and keepers are used instead of brass ones.

92. The crankpins, being the journals most liable to heat either from pounding or from friction caused by the brasses

being set up too tightly, and on account of the comparatively small surface over which the friction is distributed, require the greatest care and need constant watching. The oiling device should be of the best and the oil should never be permitted to stop feeding or the oiling device to get out of order, else there will be trouble.

LOOSE THRUST BEARING.

93. In engines fitted with some types of friction couplings, there is a thrust exerted upon the shaft in the direction of its length. This will necessitate having a **thrust bearing,** or **thrust block,** as it is sometimes called. There is a variety of thrust bearings, but the most common is the collar thrust, which consists of a series of collars on the shaft that fit in corresponding depressions in the bearing. If these collars do not fit in the depressions rather snugly, the shaft will have end play and there probably will be more or less pounding or backlash at every change of load on the engine. This can only be remedied by putting in a new thrust bearing and making a better fit with the shaft collars, unless the rings in the bearing are adjustable, as is sometimes the case, when, of course, the end play may be taken up by adjusting the rings.

WATER IN CYLINDER.

94. Pounding often occurs in the cylinders and is frequently caused by water, due to condensation or carried over from the boilers. This may be a warning that priming is likely to occur in the boilers or has already commenced. The first thing to do at such a time, if the cylinders are not fitted with automatic relief valves, is to open the drain cocks as quickly as possible and to close down the throttle a little to check the priming.

If boilers show a chronic tendency to prime, it is because they are too small for the engine, or they have not steam space enough, or the water may be carried too high

in them, which will cause a considerable reduction of the steam space. Unsteady firing, producing great fluctuations in the steam pressure, will also cause both foaming and priming, the result of which is that water will be carried over from the boilers into the cylinders. This is always a source of danger.

Water is non-compressible; therefore, after the clearance space of the cylinder is filled and more water is allowed to enter, if there is no way for it to escape, either the cylinder head will be blown out or the piston broken. Partly closing the stop-valve of the boiler showing a tendency to prime, thereby wiredrawing the steam a little, will generally check priming, if the remedy is applied before the priming becomes violent, after which it is difficult to suppress.

LOOSE PISTON.

95. Another source of pounding in the cylinder is that the piston may be loose on the rod; this is caused by the piston-rod nut or key backing off or the riveting becoming loose, permitting the piston to play back and forth on the piston rod. If due to the nut backing off, the engine should be shut down instantly on its discovery. There is very little room to spare generally between the piston-rod nut and the cylinder head; therefore, it cannot back off very far before it will strike and break the cylinder head. After the engine is stopped and the main stop-valve closed, take off the cylinder head and set up on the piston nut as tightly as possible; there is usually a socket wrench furnished with each engine expressly for this purpose.

96. Although piston-rod nuts seldom work loose and those of vertical engines are less liable to do so than others, still as a measure of safety a taper split pin should in all cases be fitted through the piston rod behind the nut or a setscrew fitted through the nut. If, on examination, this setscrew is found slack, the cause of the nut backing off is thereby explained, and it should be screwed down solid to prevent a recurrence of the trouble.

SLACK FOLLOWER PLATE.

97. A slack **piston follower plate,** or **junk ring,** as it is called by English engineers, will cause pounding in the cylinder. It seldom happens, however, that *all* the follower bolts back out at one time unless they fit very loosely in their sockets, but it is not an infrequent occurrence that one of the follower bolts works itself out altogether and swashes about the cylinder at random. This is a very dangerous condition of affairs, especially in a horizontal engine. If the bolt should get "end on" between the piston and cylinder head, which it surely will sooner or later, either the piston or the cylinder head is bound to be broken. Therefore, if there is any intimation that a follower bolt is adrift in the cylinder, shut down the engine instantly, take off the cylinder head, remove the old bolt, and put in one having a tighter fit.

BROKEN PISTON PACKING.

98. Broken packing rings and broken piston springs will cause a great noise in the cylinder, but it is more of a rattling than a pounding noise, and the sound will easily be recognized by the practiced ear. There is not so much danger of a serious breakdown from these causes as may be supposed, from the fact that the broken pieces are confined within the space between the follower plate and the piston flange. Although they rattle around in the cylinder and make a startling din, they cannot get out or do much harm, aside from causing a leaky piston in the case of the packing rings breaking or possibly slightly scoring the cylinder face. As a matter of course, this should be repaired as soon as possible.

PISTON STRIKING HEADS.

99. There is another source of pounding in the cylinder that is usually confined to old engines; it is produced by the piston striking one or the other cylinder head. One of

the causes of this is the wearing away of the connecting-rod brasses. Keying up the brasses from time to time has the effect of lengthening or shortening the connecting-rod, depending on the design, and this change in length destroys the clearance at one end of the cylinder by an equal amount. The remedy is to restore the rod to its original length by placing sheet-metal liners behind the brasses; this obviously will move the piston back or ahead and restore the clearance. It may be mentioned here that the length of a connecting-rod is measured from center to center of the bore of the crankpin and wristpin brasses.

It is obvious that the piston should never be allowed to strike the cylinder head. This condition generally is not reached suddenly; it is brought about gradually, covering a considerable period of time amply sufficient to forestall any such occurrence. A rather rare case of the piston striking the cylinder head is due to the piston rod unscrewing from the crosshead, in case it is fastened by a thread and check-nut. To obviate any danger, the check-nut should be tried frequently.

Every reciprocating steam engine should have the length of the stroke and the clearance space at each end of the cylinder marked on the guides; by this means the relative positions of the piston and cylinder heads can always be seen at a glance.

IMPROPER STEAM DISTRIBUTION.

100. The primary cause of another source of pounding is the improper setting of the steam valve, or possibly its improper design, as it cannot always be accepted as granted that the valve is properly designed. In the case of improper setting of the valve, insufficient compression, insufficient lead, cut-off too early, and late release may all cause pounding on the centers.

101. The manner in which insufficient compression causes pounding may be explained as follows: For practical

reasons there must always be some lost motion at the wrist-pin, crankpin, and shaft bearings. Now, in passing the dead centers, the direction of pressure is suddenly reversed, and in consequence the piston rod, connecting-rod, and crank-shaft will be suddenly thrown forwards by the inrushing steam to an extent depending on the lost motion at the pins and shaft bearing. It is this sudden changing of the lost motion from one brass to another, with a violence that may be likened to a blow, that causes an engine to knock in passing the centers when compression is insufficient.

102. The effect of a reversal of pressure is clearly shown in Fig. 8. With the crankpin at *a* and the engine running

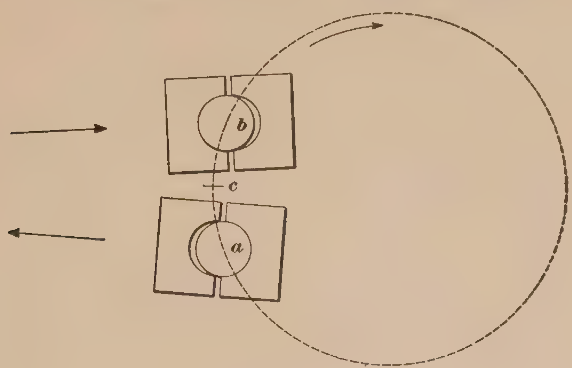


FIG. 8.

over, the connecting-rod is subjected to a pull, but after the crankpin has passed the dead center *c*, the connecting-rod is subjected to a push, in which case the rear brass, as shown at *b*, bears against the crankpin, while in the former case, as shown at *a*, the front brass bears against the crankpin.

By giving a sufficient amount of compression, the lost motion in the pins and journals is transferred gently from one side to the other before the crankpin reaches the dead center, so that by the time the live steam suddenly acts on the piston it cannot throw the rod forwards. If the compression is insufficient to gently take up the lost motion, there will be pounding.

103. Too much compression causes such a great resistance to the motion of the crank that it will tend to slow it down and thus increase the unsteadiness of the engine. Abnormal compression manifests itself by a dull, muffled sound in the cylinder or on an indicator card by the compression line rising above the steam line. It may cause pounding at the journals.

104. Insufficient lead is a common cause of pounding; in fact, it is rare to see an indicator card that shows sufficient steam lead. The exact amount of lead to be given to prevent pounding can only be determined by an actual trial; in general, slow-speed engines will require less lead than high-speed engines. In most engines the lead can be readily changed by a proper adjustment of the valve gear. In automatic cut-off, high-speed engines of the shaft-governor type, however, it is not possible, as a general rule, to change the lead by any simple adjustment, the lead having been fixed by the builder, and a change of it will require an extensive rebuilding of the governor.

105. The reason that insufficient lead causes an engine to pound is because the piston has then little or no cushion to impinge upon as it approaches the end of its stroke, and it is brought to rest with a jerk, as it were. A similar effect will be produced by a late release; the pressure is retained too long on the driving side of the piston. The ideal condition is that the pressures shall be equal on both sides of the piston at a point in its travel just in advance of the opening of the steam port. The position of this point varies with the speed of the piston and other conditions that the indicator card only can reveal; in fact, all conditions dependent on the set of the steam valve can be investigated only by the help of the indicator card. Any departure from the ideal condition above mentioned will produce more or less pounding in an engine.

106. A too early cut-off will expand the steam down too low—even below the back-pressure line sometimes.

This is an abnormal condition, which will cause pounding, and should not be permitted to occur.

A very high vacuum in a condensing engine will sometimes cause pounding by not permitting sufficient cushion for the piston to impinge upon.

POUNDING AT CROSSHEAD.

107. The crosshead is a prolific source of thumping and pounding from various causes, of which the getting loose of the piston rod is one of the most common causes. There are several methods of attaching the piston rod to the crosshead. The rod may pass through the crosshead with a shoulder or taper, or both, on one side of the crosshead and a nut on the other; or the rod may be secured to the crosshead by a cross key, instead of the nut; or the end of the rod may be threaded and screwed into the crosshead, having a check-nut to hold the rod in place. In the first-mentioned case, the nut may work loose, which would cause the crosshead to receive a violent blow, first, by the nut on one side and then by the shoulder or taper on the other at each change of motion of the piston. The remedy is obvious—set up the nut. A similar effect will be produced if the cross key should work loose and back out, the remedy for which is to drive in the key. In the case of the piston rod being screwed into the crosshead and the rod slacking back, the danger is that the piston will strike the rear cylinder head. The check-nut should be closely watched.

108. Another source of pounding at the crosshead is loose wristpin brasses, the remedy for which is to set up on the brasses, but not too tight.

109. In the case of a crosshead working between parallel guides, pounding may be caused by the crosshead being too loose between the guides; in that case the crosshead shoes should be set out.

110. In the case of a slipper crosshead, pounding will result from the wearing down of the shoe, the cure for which is to put a liner between the shoe and the foot of the crosshead or to set it out with whatever means of adjustment are provided.

POUNDING IN AIR PUMP.

111. Pounding in the air pump is generally produced by the slamming of the valves, caused by an undue amount of water in the pump, which will usually relieve itself after a few strokes. The pump piston, however, may be loose on the piston rod or the piston rod may be loose in the crosshead, either of which will cause pounding. A broken valve may also cause pounding in the air pump, all of which must be repaired as soon as detected.

POUNDING IN CIRCULATING PUMP.

112. In a circulating pump of the reciprocating type, pounding may be caused by admitting too little injection water, and the pounding may be stopped by adjusting the injection valve to admit just the right quantity of water. It may so happen, however, that the injection water is very cold, and to admit enough of it to stop the pounding in the circulating pump will make the feedwater too cold. To meet this contingency, should it arise, an air check-valve is often fitted to the circulating pump to admit air into the barrel of the pump as a cushion for the piston; this check-valve may be kept closed, when not needed to admit air, by means of a screw stem above it.

A broken valve, the piston loose on the piston rod, or the piston rod loose in the crosshead will all cause pounding in the circulating pump, the same as in the air pump, and they should all be treated in the same manner as was specified for similar troubles in the air pump.

CONCLUSION.

113. The derangements causing pounding, as well as derangements of machinery in general, produce their own individual sounds, which are easily recognized by the experienced engineer. It is here that the attentive and careful engineer will prove his value, as by taking prompt and judicious action he will prevent a breakdown. He should be able to detect any unusual noise about his engine, though it may be imperceptible to the unpracticed ear. It is almost always the case that any derangement of the parts of an engine will give timely notice by an unusual sound, and if this warning is heeded and promptly acted on by the engineer, a breakdown can generally be prevented. The various sounds produced by an engine while running can be learned only in the engine room by the engineer who is responsible for the proper running of the engine. They cannot be learned in any other way.

114. The engineer, knowing the various causes that produce pounding and thumping in his engine, can prevent them in a great measure by keeping the engine in such good order that they cannot occur.

ENGINE MANAGEMENT.

(PART 2.)

HOT BEARINGS.

CAUSE, PREVENTION, AND CURE.

GENERAL INSTRUCTIONS.

1. Introduction.—Hot bearings are the source of much anxiety and annoyance to the engineer, besides interfering very seriously with the proper performance of the engine.

2. Causes.—The primary causes that lead to the heating of bearings may be enumerated as follows

Newly fitted brasses and journals.
Refitted brasses and journals.
Brasses set up too tightly.
Brasses too loose.
Warped and cracked brasses.
Cut brasses and journals.
Imperfectly fitted brasses.
Brasses pinching the journal at their edges.
Oil feed stopped entirely.
Not enough oil.
Dirty and gritty oils, or oils of bad quality.
Oil squeezed out of the bearings.
Grit from any source in the bearings.

Journals too small, either in diameter or in length.
Overloaded engine.
Engine out of alinement.
External heat.
Brasses fitted too snugly between collars of journal.
Springing of bedplate.
Springing or shifting of pedestal or pillow-block.

3. Best Form of Bearing.—The bearing of an engine in which the shaft journals run should approximate, as nearly as possible, a hole through a solid support. If it were possible, a hole with a bushing of suitable metal in it would form the best possible bearing for a shaft; but since the bearing, however well designed and made, will in course of time wear somewhat, it becomes a necessity that there should be some means of adjusting the brasses, so as to prevent the shaft having a side movement when they are worn.

4. Adjustment of Bearings.—Some engineers consider it an error to make bearings adjustable; they say it gives an opportunity for careless men to do mischief through lack of judgment. It is certainly a fact that one of the principal causes of hot bearings is setting them up too tightly. Some persons, as soon as they hear a pound or noise about an engine immediately conclude that some bearing is slack and tighten it up; this propensity is to be deplored. There are numerous other causes of pounding in engines besides slack bearings, and the engineer should be fully convinced that the pound is caused by slack brasses before setting them up. Bearings on an engine that is in line and in good order, if properly adjusted, will run smoothly and noiselessly for months without having to be touched with hammer or wrench, and it should be the object of an engineer to get his engine into that condition as soon as possible and to keep it so.

5. Watching Bearings.—Bearings, particularly those of large engines, require constant watching. The engineer or oiler should know at all times the condition of every bearing and oil cup; this will require frequent trips around the engine to examine the oil cups to ascertain if they

are feeding and if they contain sufficient oil and to replenish the oil in the cups whenever necessary. While making his rounds, he should feel with the palm of his hand the brasses of those bearings that have shown a tendency to heat and those that are most liable to heat, particularly the crankpins.

TREATMENT OF HOT BEARINGS.

6. Mixtures for Reducing Friction.—Should any of the bearings show an inclination to heat, as indicated by its temperature rising above blood heat or above the temperature of the surrounding atmosphere, the oil feed should be increased; if the oil does not feed freely, run a wire through the oil tubes. If the bearing continues to get hotter, mix some flake graphite (black lead), flour sulphur, or powdered soapstone with the oil and feed the mixture into the bearing through the oil holes, between the brasses, or wherever else it can be forced in. A little aqua ammonia introduced into a hot bearing will sometimes check heating by converting the oil into soap by saponification, soap being an excellent lubricant. Mineral oils will not saponify.

7. Danger of Increasing Heating.—If, after trying the remedies just mentioned, the bearing continues to grow hotter, say to the extent of scorching the hand or burning the oil, it indicates that the brasses have been expanded by the heat and that they are gripping the journal harder and harder the hotter they get; at this stage, if the engine is not stopped or if the heating is not checked, the condition of the bearing will continue to grow worse as long as the engine is running, and may become so bad as to slow down and eventually stop the engine by excessive friction. By this time the brasses and journal are badly cut and in bad condition generally, and the engine must be laid up for repairs.

8. Remedies for Increasing Heating.—The state of affairs mentioned in Art. **7** should not be permitted to be reached. After the simple remedies given in Art. **6** have been tried and failed to produce the desired result, the

engine should be stopped and the cap nuts or key of the hot bearing should be slacked back and the engine allowed to stand until the bearing has cooled off. If necessity requires it, the cooling may be hastened by pouring cold water upon the bearing, though this is objectionable, as it may cause the brasses to warp or crack by unequal contraction. Putting water on a very hot bearing should be resorted to only in an emergency, that is, when an engine *must* be kept running regardless of a spoiled pair of brasses. Water may be used on a moderately hot bearing without doing very much harm. It is quite common in practice, when sprinklers are fitted to an engine, to run a light spray of water on the crankpins when they show a tendency to heat, with very beneficial results.

9. If the engine is not started again until the faulty bearing has become perfectly cool, the cap nuts or key should be set up a little, but not too much, before starting; otherwise, the brasses, having been slacked off, may be too loose, and excessive thumping and pounding will ensue.

10. Dangerous Heating.—Should a bearing become so hot as to scorch the hand or to burn oil before it is discovered or through the necessity of keeping the engine running from some cause, it is imperative that the engine should be stopped, at least long enough to loosen up the brasses, even though it is necessary to start up again immediately, otherwise the brasses will be damaged beyond repair and deep grooves cut into the journals. If the brasses are babbitted, the white metal will melt out of the bearing at this stage. The engine is now disabled, and if there is not a spare set of brasses on hand, it will be inoperative until the old brasses are rebabbitted, if they are worth it, or until a new set is made and fitted. If an attempt is made to rebabbitt a brass while it is in place under the shaft, the chances are that the attempt will result in a failure.

11. Keeping Engine With Hot Bearing Running. If it is absolutely necessary in an emergency to keep the

engine running at all hazards while a bearing is very hot, the engineer must exercise his best judgment as to how he shall proceed. After slacking off the brasses, about the best he can do is deluge the inside of the bearing with a mixture of oil and graphite, sulphur, soapstone, etc., and the outside with cold water from buckets, sprinklers, or hose, taking the chances of ruining the brasses and submitting to cutting the journal. Of course, the engine must be stopped as soon as the emergency has passed and the journal then stripped. It is to be expected that the journal will be found to be deeply grooved and the brasses cut and warped. If the brasses were babbitted, most of the white metal will have disappeared and little else but the framework of the brasses will be left. But if the brasses are made of solid composition or bronze, they can be refitted for at least temporary use or until new ones can be procured.

12. Refitting a Cut Bearing.—The wearing surfaces of the brasses and journal must be smoothed off as well as circumstances will permit; but if the grooves are very deeply cut, it will be useless to attempt to work them out entirely, and if the brasses are very much warped or badly cracked, it will be best to put in the spare ones if any are on hand. If not, the old ones must be refitted and used until a new set can be procured, which should be done as soon as possible. As for the journal, it is permanently damaged; temporary repairs can be made by smoothing down the journal and brasses; but at the first opportunity the journal should be turned in a lathe and the brasses properly refitted or be replaced with new ones.

13. After a bearing has once been heated up sufficiently to cut the brasses and journal or to warp or crack the brasses, it is afterwards constantly in danger of heating up again on the slightest provocation; and the engine is thereby rendered unreliable and uncertain in regard to its steady running. No precaution that can be taken to prevent the heating of bearings is too great to be used for the attainment of this end.

CAUSES OF HOT BEARINGS IN DETAIL.

NEWLY FITTED BRASSES AND JOURNALS.

14. Cause of Friction.—The bearings of new engines are particularly liable to heat, due to the wearing surfaces of the brasses and journal having just been machined. Newly worked metal, when viewed through a powerful microscope, presents the appearance of being a mass of fine needle points projecting outwards. When the newly worked surfaces of two pieces of metal are rubbed together under pressure, the needle points of one piece engage with the needle points of the other piece and excessive friction is produced, the result being that the surfaces in contact are cut into grooves, which still further increases the friction; but if the rubbing process is continued in a moderate manner, so that the surfaces in contact do not cut, the needle points will be bent over gradually, each point forming a small hook. Millions of these little hooks side by side form a shell or a hard surface on the rubbing parts, and the needle points can no longer engage with each other, thereby lessening very greatly the danger of heating by friction and eliminating it entirely when properly lubricated.

15. Wearing Down Bearings.—The conditions mentioned in Art. **14** exist with new brasses and the journal of an engine bearing; therefore, if a new engine or one with new brasses is run moderately, in regard to both speed and load, and with rather loose brasses, until the needle points are bent over, there will be little danger of the bearings heating thereafter from this cause if proper attention is given to their adjustment and lubrication. This is what is familiarly termed **wearing down the bearings.** The impression generally conveyed by this expression is that the metal of the brasses and journal is actually worn away; such is not the case, however, as has been explained. If the journal is true and if the brasses are properly fitted to it, there is no necessity for them to be *worn* down; to bend over the needle points is all that is required.

16. Uneven Bearing of Brasses.—Another source of heating of bearings of new engines is the following: For practical reasons there must be a little play between the brasses and their beds; this permits a slight movement of the brasses when pressure is exerted on them by the shaft; and notwithstanding the fact that they may have been most carefully fitted in the shop, they require a certain amount of running to properly adjust and accommodate themselves to their surroundings. This is especially the case with the bearings of large engines, and the same conditions will obtain every time the brasses are removed. It seems almost impossible in practice to put the brasses of a large bearing back again just where they were before removal; it always requires time for them to settle into their old places; therefore, they should not be disturbed unless there is a positive necessity for doing so. The direct cause of the tendency to heat in this instance is that the brasses do not bear evenly on the journal after the several parts of the bearing are assembled. When a bearing runs well, it is not good practice to disturb it; it is better to leave well enough alone.

REFITTED BRASSES AND JOURNALS.

17. The bearings of an engine that has just been thoroughly overhauled and the journals and brasses of which have been refitted are liable to heat. The wearing surfaces of the bearings having been newly worked or machined, the surface of the metal is in the needle stage, and, also, the brasses have not yet had a chance to adjust themselves to the journal and their beds. The engine, therefore, is in about the same condition as a new engine, so far as the bearings are concerned, and should be treated in the same manner, i. e., it should be run moderately, with loose brasses, until the needle points are bent over and a shell has been formed on the wearing surfaces, and until the brasses have accommodated themselves to their surroundings.

BRASSES SET UP TOO TIGHTLY.

18. When the brasses of an engine bearing are set up too tightly, heating is inevitable, and probably more hot bearings result from this cause than any other, and with less excuse. It is often the case that an attempt is made to stop a thump or a pound in an engine by setting up the brasses when the thump could and should be stopped in some other way.

19. The direct cause of heating of bearings when the brasses are set up too tightly is the abnormal friction that is produced by the brasses binding on the journal. The prevention and cure are obvious. The brasses should not be set up too tightly, and if they are, they should be slacked off as soon as possible. As a matter of fact, hot bearings should never occur from this cause. Only a responsible person should have charge of the bearings and no one else should be permitted to meddle with their adjustment.

BRASSES TOO LOOSE.

20. Bearings may heat on account of the brasses being too loose. The heating is caused by the hammering of the journal against the brasses when the crankpin is passing the dead centers. This derangement is easily remedied, however, by setting up the cap nuts or key. Here the experience and judgment of the engineer is called into play to decide just how much to set up, as it is very easy to overdo the matter and set up too far, with a hot bearing as the result.

21. Most practical engineers have their own particular views regarding the setting up of bearings. One method is to set up the cap nuts or key nearly solid and then slack them back half way; if the brasses are still too loose, they are set up again and slacked back less than before, repeating this operation until the ideal position is reached, that is, when there is neither thumping nor heating. It is important that this desired point be approached very gradually and

carefully, else the chances are that it will be overreached and the operation will have to be repeated all over again.

22. Another method of setting up journal brasses is as follows: Fill up the spaces between the brasses with thin metal liners, say from 18 to 22 Birmingham wire gauge in thickness, and a few paper liners for fine adjustment; put in enough of them to cause the brasses to set rather loosely on the journal when the cap nuts or keys are set up solid. Run the engine for a while in that condition and note the effect; then take out a pair of the liners and set up solid again. Repeat this operation until the brasses have reached the ideal point, when there is neither thumping nor heating, and there let them remain as long as they fill the ideal condition. It may require a week or more, and with a large engine longer, to reach the desired point, but it will be all the better to give the needle points time to be bent over and the brasses time to adjust themselves. If this system of treating bearings is carefully carried out, there will be very little danger of their heating. When the proper point is reached, the engine should run a long time without requiring any further adjustment of the bearings. In removing the liners, great care should be exercised not to disturb the brasses any more than is absolutely necessary. A pair of thin, flat-nosed pliers will be found useful in slipping out the liners. This method is preferable to the first one mentioned, because there is not so much danger of setting the brasses up too far.

WARPED AND CRACKED BRASSES.

23. Warped and cracked brasses will cause heating, because they do not bear evenly on the journal, and hence the friction is not distributed over the entire surface, as it should be. The remedy will depend on the extent of the distortion of the brasses. If the distortion is not too great, the brasses may be refitted to the journal by chipping, filing, and scraping; but if they are twisted so much that they cannot, within reasonable limits, be

refitted, nothing will do but new brasses. Warped and cracked brasses are the result of putting water on them while they are very hot, which should be avoided if possible.

CUT BRASSES AND JOURNALS.

24. Brasses and journals that have been hot enough to be cut and grooved are liable to heat up again any time on account of the undue friction produced by the roughness of the wearing surfaces. As long as the grooves in the journal are parallel and match the grooves in the brasses, the friction is not greatly increased; but if a smooth journal is placed between a set of brasses that are grooved and pressure is applied, the journal crushes the grooves in the brasses and becomes brazed or coated with brass, and then the coefficient of friction becomes very high and heating results.

The way to prevent heating from this cause is to work the grooves out of the journal and brasses by filing and scraping as soon as possible after they occur.

IMPERFECTLY FITTED BRASSES.

25. Faulty workmanship is a common cause of the heating of crankpins, wristpins, and bearings. The brasses in that case do not bear fairly or sit squarely in their beds, and while they appear all right to the eye, they may not be square in the bearing. A crankpin brass must sit squarely on the end of the connecting-rod and the rod itself must be square. If the key, when driven, forces the brasses to one side or the other and twists the strap on the rod, it will draw the brasses slantwise on the pin and make them bear the hardest on one side or the other, thus reducing the area of the wearing surfaces. The same is true of the shaft bearings. If the brasses do not bed fairly on the bottom of the pillow-block casting or do not go down evenly, without springing in any way, they will not run as they should, and heating will result. Chronic heating of bearings is almost always caused by badly fitting brasses. This is a defect that should be looked for and remedied at once, if found to exist.

BRASSES PINCHING THE JOURNAL AT THEIR EDGES.

26. Brasses, when first heated by abnormal friction, tend to expand along the surface in contact with the journal; this would open the brass and make the bore of larger diameter, if it were not prevented by the cooler part near the outside and by the bedplate itself.

If the brass has become hot quickly and excessively, the resistance to expansion produces a permanent set on the layers of metal near the journal, so that on cooling, the brass closes and grips the journal; it will then set up sufficient friction to heat again and expand sufficiently to ease itself from the journal, and so long as that temperature is maintained the journal runs easily in the bearing. This is why some bearings always run a trifle warm and will not work cool. A continuance of heating and cooling will set up a mechanical action at the middle of the brass, which must eventually end in cracking it, just as a piece of sheet metal is broken by continually bending it backwards and forwards about a certain line.

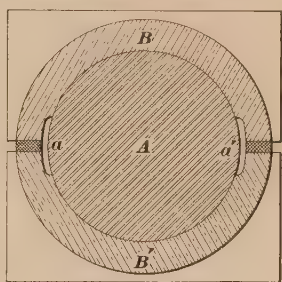


FIG. 1.

27. The cause of heating mentioned in Art. **26** may be prevented by chipping off the brasses at their edges parallel to the journal, as shown at *a* and *a'*, Fig. 1, in which *A* is a sectional view of the journal and *B*, *B* represent the top and bottom brasses.

OIL FEED STOPPED.

28. It does not take many minutes for a bearing to get very hot if it is deprived of oil. The two principal causes of a bearing becoming dry are an oil cup that has stopped feeding, either by reason of being empty or by being clogged up from dirt in the oil, and oil holes and oil grooves stopped up with accumulated dirt and gum. Both of these conditions are the direct result of negligence, and their existence can always be prevented by the exercise of reasonable care.

NOT ENOUGH OIL.

29. The effects produced upon a bearing by an insufficient oil supply is similar to that of no oil, only in a lesser degree. Of course it will take longer for a bearing to heat with insufficient oil than with none at all, and the engineer has more time in which to discover and remedy the difficulty. As a rule, however, more oil is used on bearings than is actually necessary, and a waste of oil is the result. A drop of oil at the right time and in the right place is just as good as a quart injudiciously applied. A steady feed, a drop at a time, is what a journal requires.

DIRTY AND GRITTY OILS AND OILS OF BAD QUALITY.

30. Oils containing dirt and grit or deficient in lubricating quality are prolific sources of hot bearings; but it is within the province and power of the engineer to guard against such causes. There is a great deal of dirt in lubricating oils of the average quality, as engineers find who strain it; therefore, all oil should be strained through a cloth or filtered, no matter how clear it looks. All oil cups, oil cans, and oil tubes and channels should be thoroughly cleaned out frequently. Oil may be removed from the cups by means of an oil syringe, with which every engine room should be supplied. All oil removed from the cups and cans should be strained or filtered before using. If the above instructions are strictly followed, all danger of bearings heating from the use of dirty and gritty oils will be eliminated.

31. Bearings heating from the use of oils of bad quality are not so easily disposed of, however; there is such a great variety of lubricating oils on the market whose quality cannot be definitively decided upon without an actual trial that it is a difficult matter to avoid getting a bad lot of oil sometimes. About the only safe way to meet this trouble is to pay a fair price to a reputable dealer for oil that is known

to be of good quality, unless the purchaser is an expert in oils. Cheap combination oils, generally speaking, are very deficient in lubricating qualities and hence should be avoided, as also should gummy oils, which choke up the oil channels and glue the brasses and journals together over night.

32. Brasses of very large bearings are often cored out hollow for the circulation of water through them, which assists very materially in keeping them cool.

OIL SQUEEZED OUT OF BEARINGS.

33. Bearings carrying very heavy shafts sometimes refuse to take the oil, or if they do it is squeezed out at the ends of the brasses or through the oil holes, when the journal will run dry and heat. The great weight of the shaft causes the journal to hug the bottom brass so closely that the oil cannot penetrate between them, or, if it does, it is immediately rejected. Large journals require oil of a high degree of viscosity, or heavy oil, as it is popularly called. Oil of this character has more difficulty in working its way under a heavy shaft than a thin oil has, but thin oil has not the body necessary to lubricate a large journal.

This difficulty may be met by chipping oil grooves or channels in the brasses. A round-nosed cape chisel, slightly curved, is generally used for this purpose, taking care to smooth off the burrs made by the chisel; a steel scraper or the point of a flat file will do this. The grooves are usually cut into the brass in the form of a **V** if the engine is required to run only in one direction; if it is to run in both directions, the grooves should form an **X**. In the first instance care must be taken that the **V** is forward of the direction of the rotation of the shaft; that is, the grooves should spread out from their junction in the same direction as that in which the journal turns. The oil grooves may be about $\frac{1}{4}$ inch wide and $\frac{1}{8}$ inch deep and semicircular in cross-section.

GRIT IN BEARINGS FROM ANY SOURCE.

34. Grit is an endless and ever-present source of heating of bearings; it is only by persistent effort on the part of the engineer that he can keep his machinery running cool in a dirty atmosphere. Experience is the best instructor in this matter. The causes of this condition are innumerable, therefore, it is only possible to mention a few of them here. The machinery of coal breakers, stone crushers, and kindred industries is especially liable to be affected in this way. Work done on a floor over an engine shakes dirt down upon it at some time or other; all floors over engines should be made absolutely dust-proof by laying paper between the planks to prevent this. A prolific cause of hot bearings from grit, if the engine room and firerooms communicate, is carelessness in wetting down the ashes and clinkers. If piles of red-hot clinkers and ashes are deluged with buckets of water, which is the common practice, the water is instantly converted into a large volume of steam that rises with a leap, carrying with it large quantities of small particles of ashes and grit that penetrate into every nook and cranny to which it has access, and it will find its way into the bearings sooner or later. Throwing large quantities of water on the hot clinkers and ashes should be stopped; sprinkle them instead and close the fireroom door while the ashes and clinkers are being hauled or wet down or while the fires are being cleaned or hauled.

35. If emery, emery cloth, Bath brick, or other gritty cleaning material is used around a bearing, it is sure to get inside and cause trouble; it is, therefore, better not to use them in too close proximity to a bearing.

36. As a precaution against grit getting into a bearing, all open oil holes should be plugged with wooden plugs or bits of clean cotton waste as soon as possible after the engine is stopped, and should be kept closed until ready to oil the engine again preparatory to starting up. Plaited hemp or cotton gaskets should also be laid over the crevices

between the ends of the brasses and the collars of the journals of every bearing on the engine and kept there while the engine is standing still.

37. Bearings are now in use that, it is claimed by their makers, are dust-proof, but their use does not relieve the engineer from the responsibility of taking every precaution possible to keep grit and dirt out of the bearings of his engine.

JOURNALS THAT ARE TOO SMALL.

38. Journals that have insufficient superficial area of wearing surface will heat. In practice only a certain amount of pressure *per square inch* of area can be sustained by a bearing before the friction reaches the point that will cause heating.

The pressure that a bearing will sustain *per square inch* of area of rubbing surface without heating depends on the materials of which the journal and brasses are composed, the fineness of their finish, the accuracy of their fit, the adjustment of the brasses, and the lubricant used.

39. Pressure and friction have a direct relation to each other. Less friction is produced per square inch of surface by a long journal than by a short one of equal diameter with the same total pressure; therefore, a long journal is not nearly so liable to heat as a short one of the same diameter, and a journal of large diameter is not so liable to heat as one of small diameter of equal length. It is the aim of the designer to so proportion the journal that the pressure or friction will not exceed the practical limit that the bearing will sustain. The *total* amount of friction of two bodies in contact depends on the pressure of the one on the other and is nearly independent of the area of the surfaces in contact, hence the necessity of engine journals being large enough to distribute the friction over a sufficient area of surface.

40. There is only one cure for a bearing that heats constantly on account of being too small. This is to make it

larger if circumstances permit it to be done. If this is impossible, the best of lubricant must be used, and if necessary, water must be run constantly on the bearing. It is a good idea to have a set of spare brasses in readiness for an emergency.

OVERLOADED ENGINES.

41. The effect produced by overloading an engine is precisely similar to that of the journals being too small. The pressure on the brasses being increased to a point beyond that for which they were designed, the friction exceeds the practical limit and the bearing heats. The only thing to do to remedy this difficulty is to reduce the load on the engine to within the amount it was intended to stand.

42. In the case of an engine being run at or near its limit of endurance, or if the journals are too small, especially if a large loss should be incurred by the machinery being shut down while new brasses are being made and fitted, it would be a wise and economical precaution to have a complete set of spare brasses, especially if the brasses are babbitted, on hand ready to slip in when the fatal moment arrives, as it surely will.

ENGINE OUT OF LINE.

43. If an engine is not in line, the brasses do not bear fairly upon the journals. This will reduce the area of the wearing surfaces in contact to such an extent that the friction is in excess of the practical limit, which necessarily will cause heating. If the engine is not very greatly out of line, matters may be considerably improved by refitting the brasses by filing and scraping down the parts of the brasses that bear most heavily on the journal. If this does not answer, the heating will continue until the engine is lined up.

44. The crosshead guides of an engine out of line are apt to heat, and they will continue to give trouble until the

defect is remedied. The guides may also heat from other causes; for instance, the gibs may be set up or lined up too much. Of course, if such is the case, they should be slacked off. The danger of guides heating may be very much lessened by chipping zigzag oil grooves in their wearing surfaces and by attaching to the crosshead oil wipers, made of cotton lamp wicking arranged so as to dip into oil reservoirs at each end of guides if they are horizontal, and at the lower end if they are vertical. These wipers will spread a flim of oil over the guides at every stroke of the crosshead, which will keep them well lubricated.

EXTERNAL HEAT.

45. Bearings may get hot by the application of external heat. This may be the case if the engine is placed too near furnaces or an uncovered boiler, or in an atmosphere heated by uncovered steam pipes or other means. The excessive heat of the atmosphere will then expand the brasses until they nip the journals, which will generate additional heat and cause further expansion of the brasses, and so on until a hot bearing is the result.

46. If the engine is placed close enough to a furnace to cause heating from that source, a tight partition should be put up, if possible; this will also prevent dirt and grit from the fireroom getting into the bearings. If the boilers, steam pipes, and cylinders are unclothed, they should be covered with some good non-conducting material; and possibly a ventilating fan could be rigged up to advantage. Other remedies depend on the conditions and require the judgment of the engineer.

BRASSES TOO LONG.

47. If the brasses are too long and bear against the collars of the journal when cold, they will most surely heat after the engine has been running a while; it is hardly possible to run bearings stone cold, they *will* warm up a little

and the brasses will be expanded thereby, which will cause them to bear still harder against the collars. This, in turn, will induce greater friction and more expansion of the brasses.

48. The evil may be obviated by chipping or filing a little off each end of the brasses until they cease to bear against the collars while running. A little side play is a good thing for another reason, which is that it promotes a better distribution of the oil and prevents the journal and brasses wearing into concentric parallel grooves.

SPRINGING OF BEDPLATE.

49. If the bedplate of an engine is not rigid enough to resist the vibration of the moving parts, or if it is sprung from the uneven setting or the unstability of the foundation, the engine will be thrown out of line either intermittently or permanently, and the bearings will heat from the causes and conditions mentioned in Arts. **43** and **44**; but it will do no good to refit the brasses unless the engine bed is stiffened in some way and leveled up. The form of the bedplate and the surrounding conditions generally must suggest the best way to meet this difficulty.

SPRINGING OR SHIFTING OF PEDESTAL OR PILLOW-BLOCK.

50. The effect of the springing or shifting of the pedestal or pillow-block is similar to the springing of the engine bed; that is, the bearing will be thrown out of line, with the consequent danger of heating. As the pedestal is usually adjustable, it is an easy matter to readjust it, after which the holding-down bolts should be screwed down hard. This is one of the few instances where it is permissible for the engineer to put his strength on the wrench. As a rule, a nut or bolt should be set up just solid; with very rare exceptions, a sledge hammer should never be used in driving a wrench, as 3-inch steel bolts have been broken in this way. It is also very bad practice to drive a nut up with cold

chisel and hammer, unless the nut is in a position that it is impossible to reach it with a wrench.

If a pedestal is not stiff enough to resist the strains upon it and it springs, measures should be taken to stiffen it. The method to be used can only be determined on the spot and calls for the exercise of judgment on the part of the engineer.

LUBRICANTS, LUBRICATION, AND LUBRICATORS.

LUBRICANTS.

INTRODUCTION.

51. Classification.—Lubricants may be divided into three distinct classes, viz., *animal*, *vegetable*, and *mineral*. The name of each designates its origin. There are also lubricants that are composed of a combination of two or more of the above primary classes, which practically form another, or fourth, class.

52. Properties.—The origin of lubricants is not so important to the engineer as their lubricative properties and their power to resist decomposition, vaporization, and combustion by the application of heat; especially is this the case with cylinder oils and valve oils. The value of a lubricant depends on the amount of greasy particles it contains, or its **viscosity**. Other desirable features of a good lubricant are: It should reduce friction to a minimum. It should be free from acids and free alkalies, or, in other words, it should be neutral, and of uniform constituency. It should not become gummy, rancid, or otherwise altered by exposure to the air and it should be odorless. It should stand a low temperature without solidifying or depositing solid matter. It should be entirely free from grit and all

foreign matter. It should be especially adapted to the conditions as to speed and pressure of the rubbing surfaces on which it is to be used; the question of cost is also a consideration. All first-class lubricants possess these properties to a greater or less degree, and each of them is adapted to its own particular class of work. They are also of all degrees of fluidity and solidity—from the thin, light oil used for oiling the indicator down to the thick oils and through the greases to graphite and soapstone for the heaviest journals.

53. Thick or heavy oils are generally considered to rank the highest in viscosity; this is not always the case, however. Some oils of high specific gravity rank lower in viscosity than others of a lower specific gravity, hence the lubricative qualities of an oil cannot always be judged accurately by either its viscosity or specific gravity. Then, again, different manufacturers of lubricants have different standards and names for presumedly the same grade of oil. Furthermore, lubricants that may be very satisfactory for heavy journals might not do at all for light journals, and those that answer well for journals and guides would be very objectionable in cylinders and steam chests—all of which goes to show that different lubricants are required for different purposes.

54. Selection of Lubricants.—Though there are numerous tests for determining the various properties and qualities of lubricants, they, as a rule, involve the use of elaborate chemical apparatus and complicated and delicate machines that are entirely beyond the reach of the average engineer in ordinary engine-room practice. Even the reliability of these elaborate tests is questioned and they are a source of dispute between experts. Under these circumstances it is not an easy task to instruct an inexperienced person how to select a lubricant best suited to his particular needs or to enable him to detect adulterants. Some of the simpler tests will be given further on.

55. In a general way, about the best that an engineer who is not an expert judge of lubricants can do is to procure *from a reputable dealer* several samples of oil or grease that in his judgment are best suited to the machinery he has in charge, taking care to select light-bodied oils for light machinery and to grade his selections down accordingly to suit the size and weight of the journals and the work they have to do. Then he should run the machinery for a stated length of time with each oil, carefully noting the results obtained by each. By the time the engineer has reached the end of his experiments with this assortment of oils, he will have discovered by development and observation which is the best one for his purpose, and it will then only be a matter of common sense to hold on to *that* one until he has good reasons to believe that he can get a better one; then, taking the last one as a standard, he might try another lot of *well-recommended* samples in the same way as before, and so on until he finds the best one for his purpose that the market affords, and at the same time he acquires valuable experience with lubricants. It is important, however, that he should confine his experiments to well-known standard brands of lubricants only, otherwise he will waste much valuable time without gaining a corresponding benefit, and when he finds an oil that, after a fair trial, is satisfactory, he should use it and no other.

56. In selecting the samples for trial, the engineer should examine them very carefully in every possible way and compare one with the other; he should note their color and transparency; rub some of each between the fingers and thumb or on the palm of the hand; note if the sample is smooth and oily and contains no grit; pour a few drops on a sheet of tin or a piece of glass and hold the tin or glass at different angles and note how it flows and if it leaves any residue or gum in its track; examine it with a strong magnifying glass for foreign substances; smell it, and if it is rancid or has a very offensive odor, reject it. If the engineer persists in this practice, it will not take him long to

learn how to distinguish between the different grades and qualities of lubricants, which will enable him to select the one that will best serve his purpose and, at the same time, add very greatly to his general store of engineering knowledge, thereby enhancing the value of his services

---

FLUID LUBRICANTS.

57. Animal Oils.—Animal oils are derived from the fats of animals and fish. Those that are used as lubricants are: *Lard oil*, *tallow oil*, *neatsfoot oil*, *horse-fat oil*, *sperm oil*, *whale oil*, *porpoise oil*, *seal oil*, *shark oil.*

Of animal oils, pure lard oil takes the lead as a lubricant for ordinary machinery; but it has the disadvantage of congealing in cold weather; to ward against chilling to a certain extent, the winter strained oil only should be used when it is exposed to a temperature sufficiently low to congeal the ordinary oil.

Tallow oil is very similar to lard oil.

Horse fat is sometimes used in place of tallow, but its odor is offensive.

Clarified neatsfoot oil is an excellent lubricant for light machinery.

Of the fish oils, sperm is the best lubricant, but its scarcity and high price precludes its general use. It is used, principally in a refined state, for oiling indicators and other delicate mechanisms. Whale and porpoise oils are sometimes used in place of sperm oil, but they are inferior to it. Seal and shark's liver oils are used as adulterants. Menhaden fish oil should not be used as a lubricant, as it quickly turns rancid and gums.

58. Vegetable Oils.—Vegetable oils are derived from the fruits, seeds, and nuts of trees and plants. They compare very favorably with animal oils as lubricants, and several of them are excellent for that purpose. The leading vegetable oils that are used for lubricating purposes are: *Olive oil*, *rape-seed oil*, *colza oil*, *cottonseed oil*, *castor oil*, *palm oil.*

The olive oil is probably the leading vegetable oil used for lubricating machinery, but all the others in the above list are fairly good for that purpose. Castor oil and cottonseed oils are more liable to gum than pure olive oil. Linseed oil, either raw or boiled, should not be used as a lubricant; it dries quickly and is very gummy. Cocoanut oil (palm oil) soon becomes rancid and in that condition it is not a good lubricant.

59. Mineral Oils.—Mineral lubricating oils are distilled from bituminous shale and from the residuum of crude petroleum after the volatile oils and illuminating oils have been distilled off at various temperatures up to 572° F. The products of the petroleum still, when heated to temperatures above 572° F., are the lubricating oils. These oils are graded according to their specific gravities and are named as follows:

PROPERTIES OF MINERAL OILS.

No.	Name.	Specific Gravity.	Flashing Point.	Burning Point.
1	Solar oil	.860 to .880	370° F.	435° F.
2	Mixed oil	.880 to .890		
3	Spindle oil, No. 1	.895 to .900		
4	Spindle oil, No. 2	.900 to .906	394° F.	468° F.
5	Machine oil, No. 1	.906 to .910	426° F.	487° F.
6	Machine oil, No. 2	.910 to .915	441° F. and up.	525° F. and up.
7	Cylinder oil, pale	.915 to .920		
8	Cylinder oil, dark	.920 to .950		
9	Vulcan oils	.910 to .960		

60. Besides the oils given in the table, there are many other mineral lubricating oils on sale under different names, each manufacturer naming his own product to suit himself, but the above list will serve to show the method of grading mineral machine oils in regard to their specific gravities and their flashing and burning points.

All the mineral oils given in the table, if pure, are excellent lubricants, each one being adapted to its specific purpose for light, medium, and heavy machinery and cylinders.

The color of a mineral lubricating oil is not always an indication of its purity or value. A dark-colored oil may be purer than a light-colored one; therefore, in selecting a mineral oil, too much stress should not be laid upon its color.

61. Compounded Oils.—There is a great variety of compounded oils manufactured for all sorts of purposes and at all prices. They are, generally speaking, simply made to sell without regard to merit or value as lubricants. Herein lies the danger of being defrauded in purchasing cheap oils. They are, as a rule, compounded of thin, light oils, which lack the viscosity, or body, for lubrication, and a variety of substances to produce an artificial body that adds nothing to their lubricative properties. Most, if not all, of the adulterants used for this purpose are of a gummy nature and enemies to good lubrication. If mineral oils are used as the bases of these compounded oils, they are liable to have a low flashing point, which renders them totally unfit for use in cylinders. In fact, these oils had better be entirely ignored by the engineer; but as they are made and doctored to imitate the pure standard oils, they are well calculated to deceive the unwary, as it is not an easy matter to detect the difference between them by mere inspection.

62. A trial on the engine is the best method to test the merit of a lubricant, though some simple tests, as described under the heading "Tests of Lubricants," may be made with beneficial results.

63. Economy.—Lubricants, like everything else that is exposed for barter or sale, are worth just about what is paid for them. A good article must always fetch its price and a poor article is sold cheaply.

There is no economy in buying cheap lubricants; they cost less per gallon, but it takes more gallons to do the

required work. Now that excellent oil filters are to be had, enabling the drip oil to be filtered and used over again, there is no necessity for using cheap oils.

64. Greases.—Greases are divided into three classes, viz., *compounded*, "*set*" or *axle*, *boiled* or "*cup*."

65. Compounded greases are made by mixing cheap oils with fats, paraffin, and the various waxes. They soon become rancid, in which state they are unfit for lubrication, being instead friction producers. It is hardly necessary to say that the engineer should avoid these greases, even though they are cheap.

66. Set, or **axle, greases** are mixtures of low-grade oils and fats converted into grease by the application of lime. They are cheap greases, used principally for lubricating axles of vehicles and the like, and are familiarly known as **cart grease.** These greases are unfit to use in the bearings of engines.

67. Boiled, or **cup, greases** are those that are well adapted for engine lubrication. They are produced chemically and are not simply mechanical mixtures as are the others. They are perfectly neutral and will remain so indefinitely. They are made by saponifying fats and fatty oils with lime and dissolving the soap in mineral oil.

68. Soaps made by the use of soda or potash are soluble in water, while soaps made by the use of lime are insoluble in water.

There is a series of greases in this class that are made by saponifying the fats and fatty oils by means of caustic soda; the soaps thus made are soluble in water. These greases are good lubricants if properly made, but they are apt to contain either an excess of alkali or an excess of acid; in either case they are liable to be injurious to the bearings. Free acids or alkalies may be detected by the litmus-paper test.

69. Cup and engine greases include: Nos. 1 to 4 cup greases, Nos. 1 to 3 Albany greases, sponge greases, crank-pin greases, gear greases, lubricating packing, plumbago and graphite greases.

SOLID LUBRICANTS.

70. The solid lubricants are: *graphite*, *soapstone*, *sulphur*, *mica*, *metaline*. These may properly be classed under the general head of mineral lubricants.

71. **Graphite,** called also **plumbago** and **black lead,** is used for lubrication either in the form of a powder, flaked, compressed into bushings, or by being mixed with wood fiber and solidified in molds by pressure; this latter is called **fiber graphite.** After being removed from the molds, the forms are thoroughly dried and then saturated with a drying oil, after which they are exposed to a current of hot dry air to oxidize the oil and to harden the mass. When hard they may be worked the same as metal. Fiber graphite is claimed to be self-lubricating.

72. The powdered and flake graphite are used to mix with greases for heavy journals and also to mix with the ordinary engine oils to cool a hot bearing. When graphite is used as a lubricant, the journal becomes covered with a thin coating of graphite, which reduces friction to a minimum.

73. **Soapstone, sulphur,** and **mica,** in the form of powder, are sometimes mixed with oils and greases to improve their lubricating qualities for heavy and hot journals. Sheets of mica pressed together and held firmly in a casing have been used instead of brasses with fair success.

74. **Metaline** consists of small cylinders of graphite fitted into holes drilled in the surface of the bearing; it is said to require no other lubrication.

LUBRICATION.

75. The object of lubricating the bearings of an engine is to reduce the friction of those parts that rub against one another to a minimum and to prevent the rubbing surfaces becoming hot, which, if the rubbing is continued without lubrication, will ultimately cause seizing, thereby permanently damaging the bearings and rendering the engine inoperative. The lubricant attains its object by interposing itself in the form of a thin film between the rubbing surfaces, either by gravity or pressure, and thus prevents the rubbing surfaces coming into direct contact with one another.

76. Animal and vegetables oils have been used as lubricants for many years, but since the introduction of multiple-expansion engines and high steam pressures, mineral oils have come into very general use, especially for lubricating pistons and slide valves, for the reason that mineral lubricating oils are not carbonized by high-pressure steam as readily as are animal or vegetable oils. Moreover, animal and vegetable oils (called **fatty** oils to distinguish them from mineral, or **hydrocarbon,** oils) are decomposed by the great heat of high-pressure steam and form stearic, palmetic, and oleic acids. These acids when hot readily attack iron, steel, copper, and its alloys; therefore, cylinders, pistons, etc. are eaten away when fatty oils are used for lubricating them.

77. The acids formed by the decomposition of fatty oils are particularly destructive to steam boilers when the exhaust steam is condensed and used as feedwater, as is the case with condensing engines having surface condensers. On the other hand, mineral oils are not affected by alkalies, therefore the old method of saponifying the grease in boilers and surface condensers by boiling them out with soda or potash is ineffectual when mineral oils are used in the cylinders; in that case, if the boiler tubes or condenser tubes become coated with grease, it must be removed by hand. It is far better, however, to keep the grease out of the condenser and boilers entirely by placing an efficient grease extractor in the

exhaust pipe between the low-pressure cylinder and the condenser.

78. High-grade cylinder oils only should be used for lubricating pistons and slide valves, and the flashing point should not be lower than 400° F. The higher the temperature of a hot bearing, the less is the lubricating power of the oil or grease used; consequently, a lubricant that may be thoroughly efficient at ordinary temperatures may be ineffectual in reducing the friction of a bearing that has suddenly become heated; hence the practice of mixing graphite, flour sulphur, etc. with the oil to increase its body and lubricative properties.

TESTS OF LUBRICANTS.

APPARATUS.

79. In giving the following simple tests of lubricants, there is no pretence that they are absolutely accurate in the sense of determining the commercial values of lubricating oils and greases, but they will serve the purposes of the engineer very well and assist him greatly in the selection of his lubricants; besides, they have the merit of being within the reach of every engineer in his ordinary engine-room practice.

80. The pieces of apparatus required to make these tests are few and inexpensive. They consist of an ordinary tin or iron pan 8 or 10 inches in diameter and 3 or 4 inches deep; a metal cup about the size and shape of an ordinary tumbler; a high-grade thermometer that will measure at least 500° F.; a couple of quarts of clean white sand; a half-dozen clear, white glass ½-pint bottles; a large sheet of tin or plate of glass, preferably the latter; a sheet each of red and blue litmus paper; a common thermometer; a quart of gasoline; a few pounds of ice; a pound or two of rock salt and about the same quantity of sal soda (washing soda); a small iron boiler or saucepan; a small quantity of caustic soda or concentrated lye; a pane of glass painted black on one side with

a mixture of shellac varnish and lampblack; and a small tin funnel.

81. Test for Acids and Alkalies.—Dissolve a small quantity, say a teaspoonful, of the oil or grease to be tested in five or six times its bulk of boiling water, in which steep a piece of red litmus paper; if the litmus paper remains red after having been soaked in the mixture for a considerable length of time, the oil or grease is *acid*. If the color of the paper turns to dark blue quickly, the oil is *alkali*. If it changes color very gradually to a light blue, the oil is *neutral*. As a check on the above test, try the mixture with a piece of blue litmus paper in the same way. If the color of the paper does not change, but remains dark blue, the oil is alkali. If the paper turns red quickly, the mixture is acid; but if the paper changes very gradually to a pale red, the solution is neutral.

82. Test for Viscosity. — Pour a few drops of each sample of oil upon the large sheet of tin or glass while the sheet is perfectly level, then raise one end of the sheet gently about 1 inch and support it in that position; watch the race of the drops of oil down the inclined plane. The oil that reaches the bottom of the plane *last* ranks highest in viscosity. Of course, this is only a comparative test, but it will enable the operator to select the oil best adapted to his purpose from a number of samples. After making a selection, it would be well to try the precipitation test given in Art. **87** on it for artificial viscosity.

83. Greases cannot be tested for viscosity in the way described in Art. **82**; about the only convenient method for the engineer to do this is by rubbing some of the grease between his fingers and thumb or in the palm of the hand, noting the result. After some practice he will be able to judge approximately the viscosity of the sample.

84. Flashing and Burning Tests.—Pour some of the oil that is to be tested into the metal cup until it is nearly full; place the cup in the pan and surround the cup with

sand until the pan is filled with it; place the pan and contents on a hot stove, over a gas jet, or in any other convenient place for heating it; immerse the bulb end of the high-grade thermometer in the oil in the cup and watch the rise in temperature; when it reaches 300° pass a lighted match slowly across the top of the cup; repeat this every two or three degrees rise in temperature until the vapor arising from the oil ignites with a flash, then note the temperature as indicated by the thermometer; it is the **flashing point.** Continue the test until the oil ignites and burns on the surface. When that occurs the reading of the thermometer gives the **burning point.**

85. The Cold Test.—Partly fill the metal cup with a sample of oil; place the cup in the pan; fill the pan around the cup with cracked ice mixed with rock salt and sal soda; cover the apparatus over with a piece of bagging or blanket and keep it covered until the oil in the cup is congealed; then remove the freezing mixture from the pan and fill the pan with hot water; when the oil in the cup commences to melt, immerse the bulb of a thermometer into it and note the temperature; it is the **congealing point.**

86. Saponification Test.—If it is desired to ascertain if animal or vegetable oils are mixed with oil that is represented to be pure mineral oil, it may be determined as follows: Place about a pint of the oil into the small iron boiler or saucepan and add 1 or 2 ounces of caustic soda or concentrated lye; boil the mixture for $\frac{1}{2}$ hour and then set it aside to cool. A tablespoonful of chloride of sodium (common salt) thrown into the mixture while cooling will hasten the process. When thoroughly cool, examine the mixture; if the surface is covered with soap, the oil contains animal or vegetable fats; otherwise it is pure mineral oil.

87. Precipitation Test.—The precipitation test is for the purpose of ascertaining if the oil contains paraffin, waxes, gums, etc. Place an ounce of each of the oils in a separate $\frac{1}{2}$-pint bottle, pour 2 ounces of gasoline into each bottle on top of the oil, and shake the bottles until the oil is dissolved

by the gasoline; then allow the mixtures to settle. If there is any considerable amount of precipitation or sediment in any of the bottles, it indicates that the oil in them has been treated to produce artificial viscosity and should be rejected.

88. Test for Mineral Oil Mixed With Fatty Oils.—The presence of mineral oil when mixed with animal or vegetable oils may be detected by pouring a drop of the suspected oil upon the sheet of blackened glass and holding the glass at various angles to the light; if it shows rainbow colors, it contains mineral oil.

89. Test to Detect Sulphur in Mineral Oils.—Heat a small portion of the oil to 300° F. in the metal cup and pan of sand and maintain that temperature for about 15 minutes; after cooling, if the sample is considerably darker in color than the original oil, it is unfit to use in cylinders or on hot bearings.

AUTOMATIC LUBRICATORS.

CLASSIFICATION.

90. The devices used for the automatic lubrication of steam engines and similar machinery may, in accordance with their purpose, be divided into two general classes, *bearing lubricators* and *steam lubricators.*

91. A **bearing lubricator** may be defined as one intended for, and only applicable to, the lubrication of bearings. This class is divided into three subclasses, *plain* and *sight-feed* lubricators, and *grease cups.* **Plain** and **sight-feed** bearing lubricators are intended and can only be used for oil; **grease cups,** as implied by the name, are built to use grease.

92. Steam lubricators are intended for the lubrication of the moving parts in contact with the steam; they may be

divided into *mechanical*, *water-displacement*, and *hydrostatic* lubricators. A **mechanical steam lubricator** generally has the form of a force pump; it may be operated by hand, in which case its action is intermittent. A hand-operated mechanical steam lubricator is generally fitted only as an emergency device, to be used when the automatic lubricator is out of order. When a mechanical lubricator is operated continuously by some moving part of the engine, its action is automatic. **Water-displacement lubricators** depend for their action on condensation of steam in the reservoir containing the oil; the latter being lighter than water floats on top and overflows into a suitable passage as the water in the bottom of the reservoir increases. **Hydrostatic lubricators** depend for their operation on the pressure generated by a head of water furnished by condensation of steam.

BEARING LUBRICATORS.

93. A **plain lubricator** is the simplest form of a device for automatic lubrication; it generally takes the form shown in Fig. 2. It consists of a body *a* fitted with a central tube *b* and a removable cover *c*. The oil contained in the body *a* is led into the central tube by capillary attraction, a few strands of lamp wick, as *d*, carrying the oil over. The advantage of this oiling device is its simplicity; the disadvantages are its unreliability and its lack of adjustment of the oil feed. The latter can be adjusted to some degree by changing the number of strands of lamp wick; as the flow of oil is not in plain sight, however, there is always some doubt about the action of the lubricator.

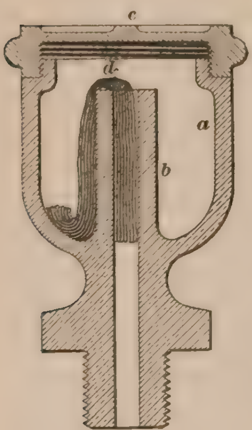


FIG. 2.

94. **A sight-feed bearing lubricator,** as implied by the name, has the oil feed in plain sight. The oil generally is fed by gravity, flowing through an annular opening in the base of the lubricator. The general appearance of this device is shown in Fig. 3. It consists of a glass oil reservoir *a* having a central tube *b* with a valve seat inside of it and at its lower end. A valve *c*, which can be locked in any position by the locknut *d*, serves to regulate the flow of the oil. The oil enters through the hole shown in the lower end of the tube *b*. The drops of oil issuing from the tube *b* show plainly in the sight-feed glass *e*. The upper cover has a hole in it through which the reservoir is filled; a movable cover *f* serves to keep out the dust.

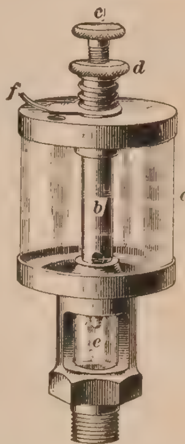


FIG. 3.

95. Various attachments are used for conveying the oil from a stationary sight-feed lubricator to the moving parts, as the crankpin, eccentric, and wristpin. Fig. 4 shows how the oil may be carried to the crankpin by a so-called centrifugal oiling device. The oil from the lubricator *a* flows through the pipe *b* into the ring *c*, which connects to a hole drilled in the center of the crankpin through the fixture *d* that is fastened to the crankpin. The oil entering at *c* passes to the crankpin by the centrifugal force generated by the revolution of the crank and through radial holes out of the crankpin between the surface of the crankpin and the brasses. The main bearing simply carries the stationary lubricator *e*, which discharges directly into the bearing. A separate lubricator *f* may be fitted for the eccentric, discharging into a long trough or funnel *g* fastened to the eccentric strap.

96. Fig. 5 will serve as a suggestion of how automatic lubrication of the wristpin and guides may be obtained. To lubricate the upper guide, the stationary lubricator *a* is used;

a lubricator *b* is placed at a sufficient distance above the level of the lower guide to cause the oil to flow through the channels shown to the guide. To lubricate the wristpin from a stationary cup *c*, a wiping device *d* is attached to the wristpin. This carries the wiper *e*, which is adjusted so as

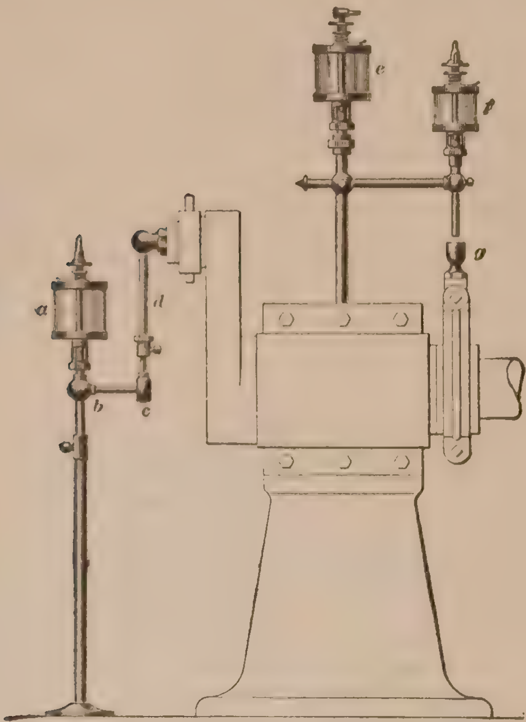


FIG. 4.

to wipe off the drops of oil hanging at the bottom of the nozzle *f* as the crosshead passes back and forth. The oil thus collected flows by gravity through a hole in the center of the wristpin and is delivered through one or more radial holes to the outside of the pin.

A precisely similar wiping fixture may be and often is used for crankpins and eccentrics, using stationary lubricators placed on top of the main bearing.

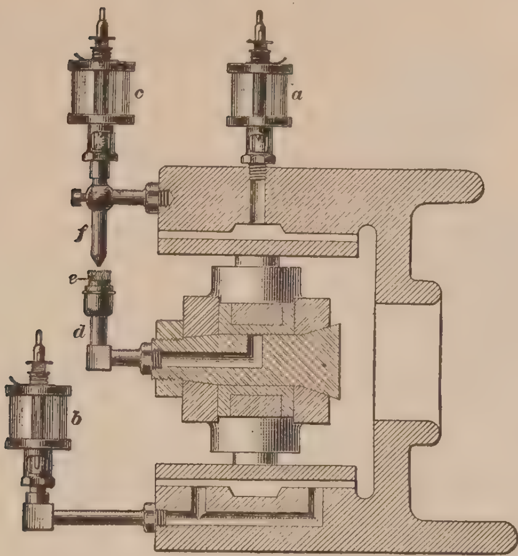


FIG. 5.

97. Grease cups are made in various ways and either as **plain** or **compression** cups. In a plain cup the grease only flows down by gravity as the heat of the bearing melts it; to assist the grease, it is a good practice to put a piece of small copper wire in the hole through which the grease leaves the cup. A compression grease cup may be hand-operated or spring-operated; Fig. 6 shows one of the type first named. By screwing the cap down by hand over the base, the grease is forced out.

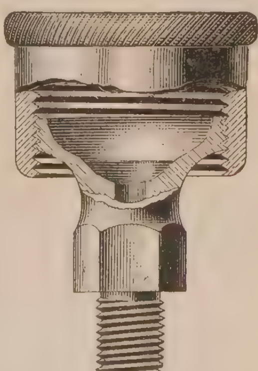


FIG. 6.

98. Spring-operated compression grease cups have a piston, on top of which is placed a spring that continually forces out the grease. In most of them the rate of flow can be regulated by a suitable valve.

STEAM LUBRICATORS.

99. Mechanical Lubricators.—Hand-operated mechanical steam lubricators are generally small force pumps connected to a suitable oil reservoir and having the discharge pipe connected to the main steam pipe close to the throttle. Their construction and operation is so simple as to require no description.

100. Automatic mechanical lubricators are operated from some moving part of the engine, as some convenient part of the valve gear. Fig. 7 shows one form of such a device, known as the "Rochester automatic lubricator." It consists of a cylindrical oil reservoir *a* containing a piston that is screwed down on the oil by gearing connected to the ratchet wheel *b*. The ratchet wheel is operated by a pawl on one end of the ratchet lever *c*, which is vibrated back and forth by the rocker *d*. This rocker is rocked back and forth by some convenient reciprocating part of the engine. The connection between *c* and *d* is made in such a manner that the arc through which *c* vibrates can be changed so that the pawl will move the ratchet wheel any desired number of teeth within the range of the device. The oil is ejected from the reservoir by the piston and passes through *e* to the engine.

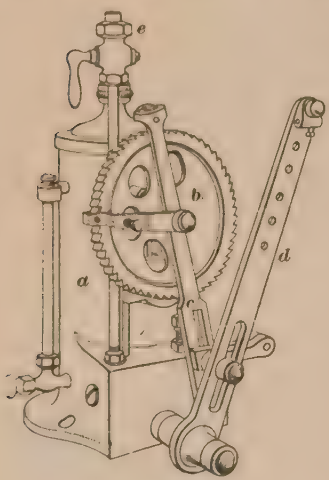


FIG. 7.

101. Water-Displacement Lubricators.—The simplest form of a water-displacement lubricator is shown in Fig. 8. It consists of a cylindrical shell *A* provided with a central tube *a*; a cap *C*, through which the lubricator is filled; and a shank *b* for attaching it in a vertical position

to the steam chest or steam pipe. A valve *B* controls the communication between the lubricator and the engine.

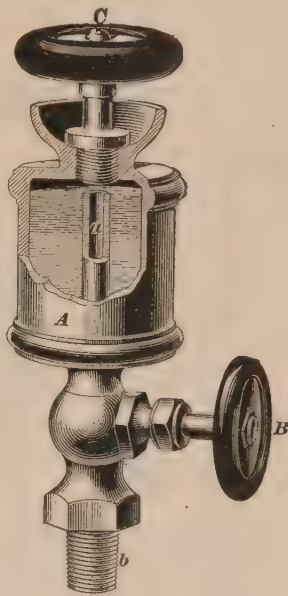


FIG. 8.

102. The operation of the lubricator is as follows: The receptacle is filled with oil and closed. The valve *B* is then opened, thus allowing the steam to pass through the central tube in to the top of the lubricator. The steam, coming in contact with the cold surfaces of the oil and receptacle, condenses. Since water is heavier than oil, bulk for bulk, the drops of condensed steam sink to the bottom of the receptacle. As two bodies cannot occupy the same space at the same time, the drops of water displace a quantity of oil equal in volume to their own; the oil, which has no other means of egress, flows over the edges of the central tube and runs by gravity into the steam pipe.

The objectionable features of this lubricator are that the flow of oil is not readily controlled and that there is no indication of when the lubricator stops working, either for want of oil or otherwise.

103. To overcome the objections mentioned in Art. **102**, **sight-feed** water-displacement lubricators have been designed, one of which is shown in Fig. 9. Its principle of action is the same as that of the lubricator shown in Fig. 8; i. e., it depends on the condensation of the steam and the subsequent displacement of the oil. Its construction is as follows: A cylindrical receptacle *d* is provided with a central tube *a* communicating with the threaded shank *e* and the sight-feed glass *A*. To fill the receptacle,

the cap *E* is provided. The upper end of the lubricator communicates with the sight-feed glass by the passage *b*. In operation the steam is admitted to the lubricator by means of the valve *B*, the opening of which admits it to the inside of the lubricator as well as to the sight-feed glass *A*. The steam, coming in contact with the oil and the top of the lubricator, condenses and displaces the oil, which then flows through the passage *b* into a conical nozzle, as shown, and issues from the latter either drop by drop or in a thin stream, depending on the position of the regulating valve *C*. It is apparent that by screwing down the latter, the annular opening between the valve and nozzle is reduced, and hence the flow of oil is checked. Conversely, by screwing up the valve, the rate of flow is increased. The drops of oil issuing from the nozzle flow by gravity through the passage *c* and thus to their destination. Since the glass tube is transparent, the oil dropping from the nozzle is in plain sight of the attendant. By means of a drain cock, not shown in the figure, the lubricator may be emptied when required. This lubricator uses a **down feed**, which means that the oil is discharged downwards in respect to the feed nozzle. These lubricators are not very reliable in their action, since the oil is not forced through the feed nozzle, but only flows through it by gravity.

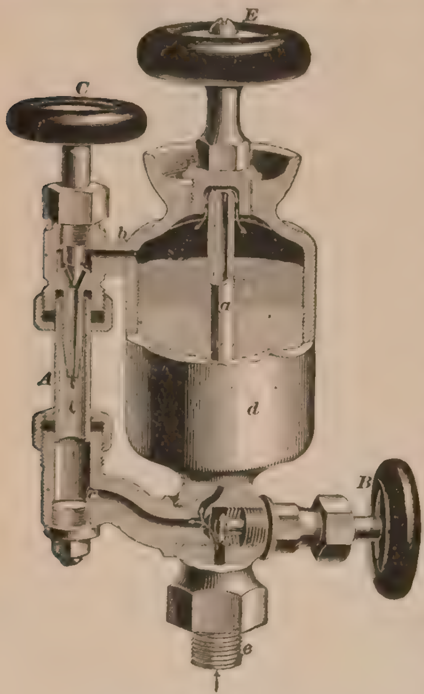


FIG. 9.

104. Hydrostatic Lubricators.—All water-displacement lubricators belong to the **single-connection** type, this meaning that there is only one connection to the steam pipe and, consequently, that the oil must pass through the same passage through which the steam is admitted. Hydrostatic lubricators are made in two styles, *single-connection* and *double-connection*. In a **double-connection** lubricator there are two connecting pipes to the steam pipe, the steam being admitted through one pipe and the oil leaving the lubricator through the other.

105. A typical single-connection hydrostatic lubricator is shown in Fig. 10, (*a*) being a part section and (*b*) a side view. The lubricator is connected to the steam pipe through the nipple *M*. The steam flows through *M* and the pipe *B* into the **condenser** *F*; it also flows through the connection *b* and a passage cored out in *C* to the sight-feed glass *H*. The steam is condensed, both in the condenser and in the sight-feed glass, by radiation. The water in the condenser flows through the pipe *1* into the bottom of the oil reservoir and forces the oil to the top, exerting a hydrostatic pressure on the bottom of the oil, which is transmitted through the oil. The latter flows through the pipe *2* into a nozzle located in the bottom of the sight-feed glass and out of the nozzle into the glass. The drops of oil ascend, by reason of oil being lighter than water, to the top of the sight-feed glass, which, it will be remembered, is filled with water. The oil then flows into the passage within *C* and passes through *b* into the nipple *M* and into the steam pipe.

106. There is an equal steam pressure on top of the water in the condenser and in the sight-feed glass, so that the pressure impelling the oil out of the lubricator is only that due to the hydrostatic head. The rate of flow of the oil through the nozzle in the bottom of the sight-feed glass can be regulated by means of the needle valve *E*; the water can be shut off from the oil reservoir *A* by closing the valve *D*; a drain cock *G* is used for draining the reservoir. A gauge glass *g* shows the amount of oil in the lubricator.

The reservoir can be filled when the filling plug O is unscrewed. A small valve S is closed in order to shut off the steam from the sight-feed glass in case the latter is broken or in need of cleaning. With the valves D and S shut, the gland in which the valve E works is unscrewed;

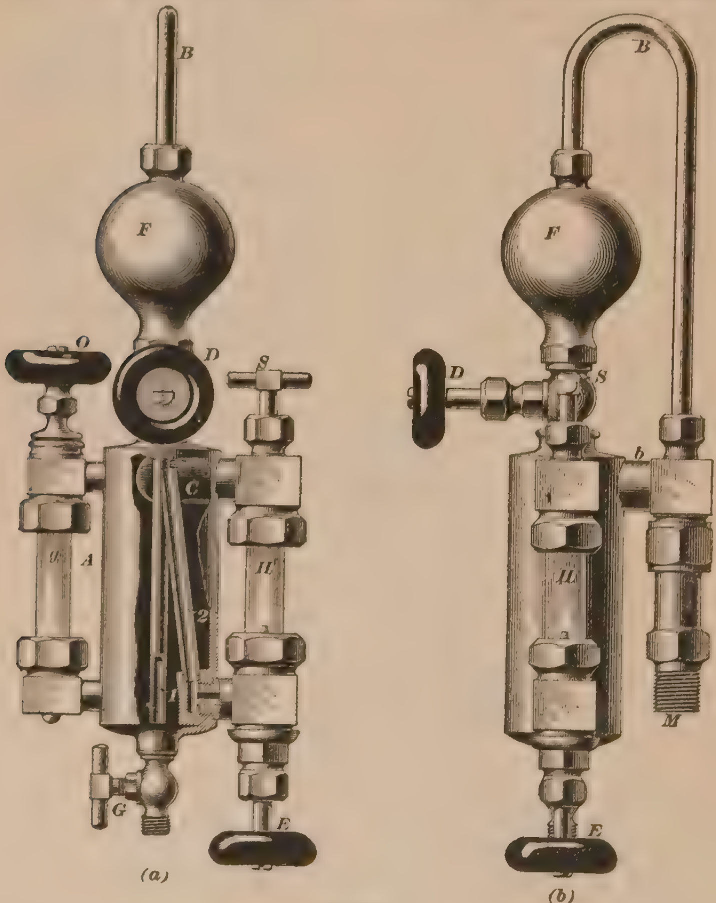


FIG. 10.

the broken or dirty glass tube can be removed and a new one or the cleaned old tube inserted. It will be noticed that this lubricator has an **up feed**; that is, the drops of oil coming from the nozzle flow upwards in the sight-feed glass.

107. To start the lubricator, open the valves *D* and *S*; to stop it, close the valves *D* and *S*. The regulating valve *E* when once adjusted need rarely be disturbed. To drain the lubricator while steam is on the pipe into which it delivers, close the valve *D*, and *E* and *S* being open, open *G*. When not under steam, to drain remove the filler plug *O* and open *G*.

108. A **double-connection lubricator** is shown in Fig. 11, which incidentally shows the mode of attachment to a vertical pipe. The condenser *a* has its independent steam connection; the angle valve *b* admits the steam to the condenser. The water in the condenser passes to the bottom of the reservoir *c* through the pipe *d* and forces the oil upwards into the pipe *e* leading to the bottom of the sight-feed glass *f*. It then flows through an annular opening regulated by the valve *g* up the sight-feed glass and through the pipe *h* and valve *i* into the steam pipe. The pressure impelling the oil forwards is simply the hydrostatic pressure due to the water in the condenser.

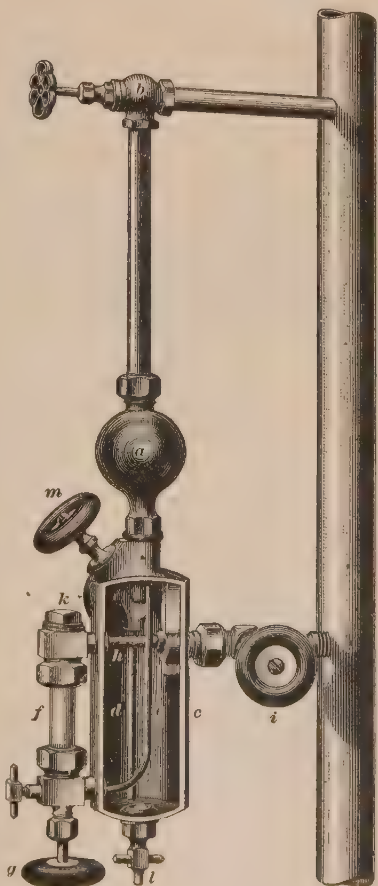


FIG. 11.

109. To fill the lubricator, close the valve *k*, which shuts the condenser off from the reservoir *c*, and also close the valve *g*. Open drain valve *l* and

remove filler plug *m*. When water has drained off, close valve *l*, fill with oil, and replace the filler plug. Open valve *k* again and regulate the flow with the valve *g*. To shut off the lubricator temporarily, close the valve *k*; to shut it off permanently, close valves *b* and *i*.

110. Double-connection lubricators are made in which the condenser is an independent vessel; such a one is shown connected up to a horizontal overhead steam pipe in Fig. 12. The receptacle *o*, which may be filled by unscrewing the cap *m*, communicates with the sight-feed glass *c*. The regulating valve *i* controls the flow of the oil. At *j* a valve used for draining the receptacle is shown; the drain pipe may be attached by the union *k*. The valve *l* may be used for closing the passage leading from the bottom of the lubricator to the pipe *a*. The lubricator is connected to the steam pipe *n* by the pipe *a*, which connects *o* to the condenser *p*, which is, in turn, connected to *n* by the pipe *f*. The oil from the lubricator passes to the steam pipe through the pipe *b*. By means of the valves *g* and *h*, the lubricator may be shut off when desired. Its operation is as follows: When starting the cup for the first time, the pipes *a* and *b* and the sight-feed glass *c* are filled with water, the pipe *a* being filled nearly up to *d*. Since the water in the pipe *a* can flow into the bottom of the lubricator, it follows that the oil will

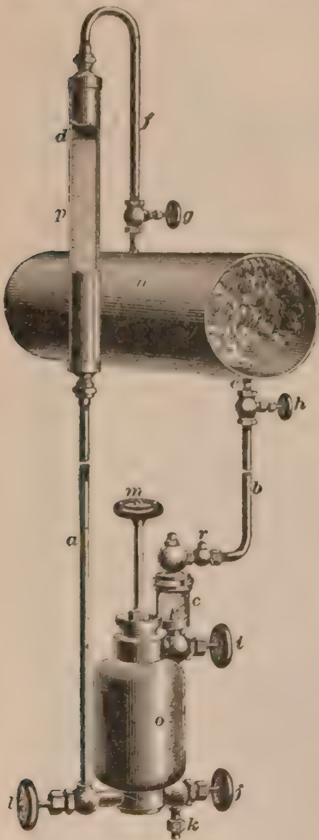


FIG. 12.

be forced through the feed nozzle with a pressure depending on the hydrostatic head *de*.

After passing through the feed nozzle, the drops of oil ascend through the sight-feed glass and up the pipe *b*, the pressure causing the upward flow being due to the difference in specific gravities of the water and oil. To prevent the emptying of the pipe *b* when draining the lubricator preparatory to replenishing the oil supply, a small check-valve *r* is provided. In order to replenish the water that passes from the pipe *a* into the lubricator, the condenser *p* is used. This may be a vessel of any desired shape; it is usually a piece of 1½-inch brass tubing, as shown in the figure. The steam entering from the steam pipe is condensed by coming into contact with the relatively cool surfaces of the condenser; the latter is made large in order to increase the radiating surface. In this style of lubricator the hydrostatic pressure operating the device may be made as great as circumstances will permit by simply extending the loop of the pipe *f* higher up. If this is done, the condenser must also be raised in order to derive the most benefit from the change.

111. Double-connection lubricators should never have one connection attached to the steam pipe between the throttle and boiler and the other between the throttle and engine. If the lubricator is connected in this manner, upon closing the throttle there will be full steam pressure on the condenser and none on the sight-feed glass. In consequence, the lubricator will very rapidly be emptied, the steam pressure forcing all the oil out into the engine. If circumstances require the connection to be made in this manner, a special locomotive double-connection sight-feed lubricator should be selected. Such a lubricator is especially made in such a manner that the oil cannot leave the reservoir when the throttle is closed.

ENGINE INSTALLATION.

COMPARISON OF TYPES OF RECIPROCATING ENGINES.

VERTICAL VERSUS HORIZONTAL ENGINES.

1. Uses of Vertical Engines.—The inverted vertical engine—that is, the engine with the crank-shaft resting in a bedplate placed on the foundation and suitable and appropriate housings containing the guides, the cylinder resting upon and secured to the housings or engine frame—is now the prevailing type for large power-station purposes and many other applications of steam engines. That type of vertical engine in which the cylinder joins the bedplate and has the shaft or beam on top of the engine framing is only used for special and peculiar applications, such as slow-speed pumping and blowing engines, but never for quick-running engines.

2. Controlling Features.—The vertical engine has its distinct application, its advantages, and its disadvantages. The two controlling features that dictate the use of the vertical engine are (1) available floor space for the engine and (2) size of engine. The first reason is self-evident. As to the second reason, in very large horizontal engines, and particularly with the low-pressure cylinder of compound engines, the problem of supporting the weight and preventing the cutting of massive low-pressure pistons running at

the speed now common becomes one of great magnitude; in fact, the success is always problematical, even with the most carefully planned constructions. This bad feature of the horizontal engine is entirely overcome by making the engine vertical; the weight of the piston is then borne by the shaft bearings.

3. Supporting Pistons of Horizontal Engines.—Many devices have been tried to support the weight of large pistons, such as tailrods having crossheads running on external guides, but the distance between the points of support or crossheads is usually long, and the allowable deflection can rarely exceed $\frac{1}{32}$ inch, so that this expedient, to be of any service whatever, requires very large rods. Take the case of a 72-inch cylinder having a 72-inch stroke. With a carefully designed cast-iron piston, it would require a piston rod at least 14 inches in diameter having a 5-inch hole through it to support the piston successfully. Pistons having a steam pocket underneath them, into which steam is admitted through a small hollow tailrod, have been used by one very large builder. Forged steel-plate pistons having broad composition shoes riveted to the lower circumference, the shoes projecting into recesses formed in the heads, have been used by an English builder; while very broad pistons in which the weight of the piston does not exceed 3 pounds per square inch of projected area are often resorted to. Many of the devices for supporting pistons have merit, but many engineers believe the vertical engine to be the best solution of the problem.

4. Inaccessibility of Vertical Engines.—The vertical engine is much more inaccessible than the horizontal machine for oiling, inspecting, and repairing; indeed, in some of the very latest American high grade engine designs, it would be necessary to dismantle the whole machine to remove the crank-shaft, although the specifications usually demand that it be possible to remove the crank-shaft bearings when the shaft is raised $\frac{1}{4}$ inch. The vertical engine costs on an average about 12 per cent. more than the

horizontal engine. Generally, the vertical engine will not receive the same degree of care and attention that the horizontal machine will, owing to its inaccessibility and the labor and exertion required to reach its various parts. This should not be the case, but it is so, nevertheless.

5. Comparison of Headroom.—The vertical engine requires quite a high building, not only on account of the design, but also because extra room is needed to draw out the piston and piston rod. If the engine is large, a substantial crane or other means of handling the various parts is necessary. The horizontal engine, where space is available and other conditions do not preclude its adoption, has many practical and commercial advantages over the vertical engine. It is the cheaper engine, is much more accessible for repairs, oiling, and inspection, and can be cared for by men physically incapacitated to handle a vertical engine.

6. Comparison of Floor Space.—The horizontal engine from the nature of its design requires considerable floor space, and in localities where property is valuable, as is often the case along city water fronts or at the center of a large electrical distributing system, and where it is desirable to concentrate as much motive power in as small a space as possible, the horizontal engine must give way to the more expensive and less accessible vertical engine. In very large power plants it is quite customary either to connect the engine galleries, making them continuous throughout the whole plant, or to construct mezzanine galleries around the house on the same level and connecting with all the galleries. If this is done, the various units can be visited for inspection and oiling without descending to the floor each time, thus making easier the labor of attending to this class of engine.

7. Influence of Drainage.—Unless especial care is exercised in the design of the vertical engine to free of water all parts coming in contact with the live steam and to prevent water pockets, its economy will fall below that of the horizontal engine, especially if the latter is of the four-valve

type, which type when embodied in the horizontal machine is almost self-draining. The vertical design does not so readily lend itself to drainage and hence requires especial care in this respect.

8. Influence of Balancing.—The mechanical efficiency of the vertical engine is from 2 to 3 per cent. higher than that of the horizontal engine. With an equal measure of care as regards balancing, the vertical engine will operate more smoothly than the horizontal machine; this is due to the fact that the unbalanced vertical force acts vertically through the machine and foundation, while the unbalanced horizontal force is close to the foundation and is counteracted by two heavy masses—the foundation below and the engine above.

9. Combined Vertical and Horizontal Engines.—A type of engine occasionally used is a combination of the horizontal and vertical machine. This engine is usually made a compound, in which the low-pressure cylinder is made vertical, for reasons that have been previously given, while the high-pressure cylinder is placed horizontal. Both engines act on one crankpin, thus making a compact machine having all the advantages of two cranks at right angles. This type of machine has been made in very large units and used for direct-connected electric service and reversing rolling-mill service.

AUTOMATIC CUT-OFF VERSUS THROTTLING ENGINES.

10. Types of Automatic Cut-Off Engines.—There are two distinct types of automatic cut-off engines, the *positive automatic cut-off*, in which the main or cut-off valve closes the port by positive motion derived from some part of the engine, usually an eccentric on the main shaft, and the *releasing-gear cut-off*, in which the main valve or cut-off valve is made to close the admission port by means of vacuum pots, weights, springs, steam pressure, or other sufficient means.

11. Limiting Speeds of Releasing-Gear Engines.—The positive automatic cut-off engine is always used for quick-running engines in preference to the releasing-gear engine. The practicable maximum speed of the latter type may be set at 100 revolutions per minute for engines up to 500 horsepower, 85 revolutions per minute up to 2,000 horsepower, and 75 revolutions per minute up to 5,000 horsepower, although small releasing-gear engines have been run at 150 revolutions per minute. The best builders do not advise speeds higher than those given above. There are a very few builders that run their small and medium-size engines about 20 per cent. faster than the speeds given above, but this does not mean that the same power is obtained at 20 per cent. less investment, for the reason that to run satisfactorily at these high speeds the machines must be especially constructed and heavily built; furthermore, the lack of insurance against shut down due to breakage or heating and the larger quantity of oil required more than offset the item of first cost. For all conditions requiring high speeds and great economy, the positive automatic cut-off engine is usually chosen. These machines can always be run at speeds not determined or limited by the construction or operation of the valve gear.

12. Economy of Automatic Cut-Off Engines.—The economy of the positive automatic cut-off engine with one valve is only about 75 per cent. of that of a releasing-gear automatic cut-off engine of equal grade. There are, however, some positive automatic cut-off engines of the four-valve type in which the cut-off valve is mounted on the main valve and is positively driven by a shaft governor; in such an engine the economy of steam is fully equal to that of the releasing-gear engine. They are capable of much higher speeds than the releasing-gear engine, but owing to some complication of the valve gear, they are not usually run at as high speeds as the one-valve positive automatic cut-off engine. The four-valve positive automatic cut-off engine is somewhat more expensive than the

releasing-gear engine, which, in turn, is considerably more expensive than the one-valve automatic engine.

13. Accessibility of Releasing-Gear Engines.—The releasing-gear engine is invariably more complicated than the positive gear and requires closer adjustment, but on the other hand, it is much more accessible for adjustment, even while in motion, than the positive gear, which is unapproachable while the engine is running. The problem of oiling the positive-gear engine is one that cannot be solved too carefully, as the success of the gear largely depends upon the perfection of the oiling devices. The oiling of the releasing-gear engine is an easy problem in comparison. These statements apply not only to the valve gear, but with equal force to the reciprocating and rotating parts of both classes of engines.

14. Comparison of Throttling and Automatic Cut-Off Engines.—The simple throttling engine is the oldest type of engine and is probably the least used at the present time. Its strongest claim to existence is simplicity, and for many purposes and locations the claim is strong. It is much less economical than the automatic cut-off engine, is usually built for slower speeds, and generally there is not much attention paid to the features conducive to economy. One of the defects of the throttling engine is that, for the purposes of regulation, this machine must have from 5 to 20 pounds more steam pressure on the inlet side of the governor than on the outlet side; consequently, the boiler must generate steam from 5 to 20 pounds higher pressure than is actually used in the engine; hence, some waste of heat takes place before the steam arrives in the working cylinder.

15. The automatic cut-off engine adjusts its energy to the resistance by measuring out a supply of steam always at or near the boiler pressure and sufficient to overcome the resistance; the throttling engine always supplies the same volume, but varies the pressure to suit the resistance to be overcome. The automatic cut-off engine is capable of

high ratios of expansion; the throttling engine as usually built is not, $\frac{3}{4}$ cut-off being the prevailing point, giving only $1\frac{1}{3}$ expansions.

16. There is another class of engine, known as the *Meyer valve engine*, belonging to the throttling-engine family, in which a separate expansion valve on the back of the main valve and worked by a separate eccentric is used to effect a cut-off from 0 to $\frac{3}{4}$ stroke. This class of engines is capable of high ratios of expansion and hence is quite economical; they are usually well made and provided with a throttling governor. The point of cut-off is adjustable by hand and is set very close to the actual demands, allowing the governor very little range of pressure to adjust the speed of the engine.

SIMPLE ENGINES VERSUS COMPOUND.

17. Influence of Power Required.—Like most engineering problems, the problems relating to the use of compound engines resolve themselves chiefly into problems of finance. The cost of fuel and amount of power required are leading factors in determining the use of compound engines, generally speaking. When the power required is less than 200 horsepower, it will hardly pay to put in a compound condensing engine where the steam pressure is limited to 100 pounds gauge, unless the cost of fuel is very high, say $4, or more, per ton. If 125 pounds of steam can be carried by the boiler, it will be a paying investment. Similar limits apply to the case of a compound non-condensing engine, except that the steam pressures should be changed to 125 pounds gauge pressure and 150 pounds, respectively.

18. As the size of the engine increases, it becomes more important to compound, for the reason that a 1,000-horsepower engine does not cost five times as much as a 200-horsepower engine of similar design and construction. When the price of fuel is low, compounding becomes of less

importance, and compound non-condensing engines with a variable load when working with steam pressure not more than 150 pounds are rarely paying investments.

19. Triple- and Quadruple-Expansion Engines.—Triple-expansion condensing engines have shown a real economical advantage over compound engines of 20 per cent. Such engines should not be used with less than 160 pounds steam pressure. Triple-expansion non-condensing engines seldom prove a good investment under ordinary conditions, and the same may be said of quadruple-expansion condensing engines. This statement refers to land engines, but not to marine engines, where quadruple engines are sometimes used not only to secure extreme economy in the use of steam, but also to reduce the vibrations of the ship to a minimum.

20. Steam Consumption.—A good four-valve automatic cut-off engine will consume 24 pounds of dry steam at 100 pounds pressure per horsepower per hour, while a compound condensing engine of similar design, but having a reheating receiver supplied with 50 square feet of tube reheating surface for each cubic foot of steam delivered from the high-pressure cylinder, will consume but 14 pounds of dry steam of 135 pounds pressure per horsepower per hour. A good triple-expansion condensing engine, if supplied with steam at 160 pounds pressure, could accomplish the same work with 11 pounds of dry steam per hour. The above figures as to steam consumption hold only for medium and large engines, say from 500 horsepower up; small engines are not so economical as large engines, which is probably due to the greater ratio of cylinder and port surface to the volume swept through by the piston in small engines.

21. Factors to be Considered.—While the problem of simple versus compound engines is always one of finance, economy of fuel and first cost are not always the determining elements. The compound engine is always more complicated and hence more liable to a breakdown, and if

isolated, requires the carrying of more spare parts. It requires a higher degree of skill to maintain it in economical condition and requires better and more expensive boilers, but does not require as many boilers or as large a boiler plant. The question of whether insurance risks may be greater and the facilities for repairs should also be considered in determining the type of engine. The real test in any case is the final influence of the machinery used on the profits of the business.

TANDEM COMPOUND VERSUS CROSS-COMPOUND ENGINES.

22. Cylinder Arrangements of Tandem Compound Engines.—A tandem compound engine is one in which the cylinders are arranged one behind the other, both pistons being on the same piston rod and acting on one crankpin. The cylinders are arranged sometimes with the high-pressure cylinder behind and sometimes with the low-pressure cylinder behind. Both arrangements have their advantages. When the low-pressure cylinder is placed behind and the front low-pressure cylinder head is made on an internal flange, the cylinders and pistons are quite accessible. When the high-pressure cylinder is placed behind the low-pressure, the piston rod must be removed through the front low-pressure stuffingbox and, consequently, can have no projecting collars forged on it to take the thrust of the low-pressure piston. The rod is sometimes fitted with loose steel collars that take the thrust of the low-pressure piston; sometimes the portion of the rod that enters the low-pressure cylinder is made rather large in order to form a sufficient shoulder for the low-pressure piston to bear against. When it becomes necessary to forge a collar on the rod to secure a sufficient bearing shoulder for the low-pressure piston, the stuffingbox throat must be bushed; the collar will then pass through the throat when the bushing is removed. Both pistons should have tapered seats on the rod.

23. Disadvantages of Tandem Compound Engines. The principal objection to the tandem engine is the inaccessibility of the cylinders and pistons for inspection or repair. The cylinders are also liable to get out of alinement if not properly designed and constructed, which occurrence, in turn, reduces the mechanical efficiency of the machine. The loss of alinement is obviated to a considerable extent by making a heavy cast-iron sole plate extend under both cylinders. The front cylinder should be securely bolted to this sole plate, while the rear cylinder should be arranged to slide in suitable ways, which constrain it laterally, but allow it to move longitudinally when it expands and contracts. This feature in large engines is important.

24. Comparison of Spare Parts Required.—The economical performance of the two types of machines are the same. The cross-compound engine has considerably more parts than the tandem, but many of them are exact duplicates, so that in isolated districts the cross-compound engine would probably not require a larger item of spare parts than the tandem. On account of their smaller size, for equal engine power, the first cost of spare parts for a cross-compound engine would be less than for the tandem engine.

25. Comparison of Mechanical Efficiency.—For equal engine power, the mechanical efficiency of the two types of machines should be in favor of the cross-compound engine. This at first thought seems erroneous, but a little consideration will make this fact clear. The frictional resistances of pistons and rods should be the same with both, but in the tandem compound they are much more liable to increase in time, due to its greater liability to get out of alinement. The valve-gear resistances should be practically the same in both types, but usually are slightly in favor of the tandem compound. Considering the resistances at the crossheads and crankpins, while there are twice as many parts in contact in the cross-compound engine, the total force and resultant pressures and the direction and duration of the pressure are the same in both types.

26. The greatest divergency in the frictional resistances of the two types of engine is at the shaft. For equal degrees of unsteadiness of rotation, both engines working at the same economical ratio of expansion and speed, the tandem engine requires a wheel about $1\frac{6}{10}$ times heavier than a cross-compound engine. This, in turn, requires a larger and heavier shaft and bearings and means an increased velocity of the bearing surfaces, and hence more wear and oil; in this respect the cross-compound engine has a decided advantage over the tandem.

27. Comparison of Cost.—The tandem engine has its strongest claim in the matter of first cost; if this, however, is carefully investigated, it will be found that for similar service, economy, speed, pressure, and type, the first cost of the tandem engine will average only about 9 per cent. lower than the first cost of the cross-compound engine. The cost of foundation for a tandem engine will be about 20 per cent. less than that for the cross-compound.

28. Formerly, it was the practice to make the passage of steam from the low-pressure cylinder to the high-pressure cylinder of tandem engines as short and direct as possible, but the prevailing practice at present for equal duty is to give the tandem engine a reheating receiver of a volume equal to that of the receiver of the cross-compound engine, which is usually equal to the volume of the low-pressure cylinder; and it is customary to provide for both types of engines about 50 square feet of tube reheating surface for each cubic foot of steam exhausted by the high-pressure cylinder. Formerly, there was considerable difference of cost between receivers and piping for tandem and cross-compound engines, but as at present constructed, there is no appreciable difference.

29. Comparison of Smoothness of Running.—With equal elaboration to secure smoothness of running, and comparing condensing engines, the tandem engine will generally excel. The reason for this is seen when it is considered that compression is the main factor tending to secure smoothness

in turning the dead centers. If the vacuum in the low-pressure cylinder be good, the remaining gas is so attenuated that ordinary means will not secure sufficient compression to absorb the inertia of the reciprocating parts at the end of the stroke, the result being a severe pounding at all journals. To prevent this, extraordinary and expensive means must be used, such as providing separate valve gear to drive the exhaust valves independent of the steam valves. In the tandem engine, both pistons being on the same rod, sufficient compression can easily be obtained behind the high-pressure and low-pressure pistons to fully absorb the inertia of the reciprocating parts. This applies particularly to releasing valve-gear engines.

SINGLE VERSUS DUPLEX ENGINES.

30. Purpose of Duplex Reversible Engines.—A duplex engine consists of two simple engines, usually exact duplicates in all respects, operating on one crank-shaft; hence, it is similar in arrangement to the cross-compound engine. Reversible engines are almost invariably duplex to facilitate starting the engines at any possible position at which the cranks may happen to be. Familiar examples of this type of engine are the locomotive, hoisting engines, blooming engines, and barring engines.

31. Purpose of Duplex Non-Reversible Engines.—The duplex non-reversible engine is frequently met with in industrial works, and their existence is usually due to an extension of the industry, where a little forethought has served to save the additional cost of an entirely new engine. In planning and developing an industry, it is reasonably expected to grow and expand; often the exact expansion cannot be predicted with certainty. While a certain amount of surplus power can be provided for in installing the original engine, it is a well-established fact in steam engineering that an underloaded engine is an extremely wasteful and poor paying investment; this fact creates the field for the

duplex non-reversible engine. The wheel for the original single engine is made sufficiently large to transmit double the original power, if belt or rope transmission is used; this, however, does not mean that it shall be double the weight or cost, but only 1.4 times the weight for a single engine and about 1.3 times the cost of a wheel for a single engine. Frequently a section of the bedplate containing the shaft bearing is purchased with the original machine, and when the demand for another engine is made, it can be readily attached to the original machine without a shut-down or delay of the works.

32. Methods of Providing for Increased Power.— Other methods are sometimes practiced to accomplish the ends secured by the duplex engine, such as purchasing a larger engine than is required, inserting a thick bushing in the cylinder, and providing a smaller piston, all being so arranged as to be removed when the demand for increased power is made. While this accomplishes the same result, it is done at a sacrifice of economy and ties up a considerable amount of extra capital, due to the first cost of the larger engine, which might otherwise be turned to good investment.

If conditions so change between the time of installing the original engine and such time as increased power is required, the machine intended to be duplex can quite as readily be made into a cross-compound as a duplex by adding a low-pressure cylinder and receiver, and if water be available, a condenser. It might be well to add here that if it is required to obtain the same amount of power with a higher degree of economy, the steam pressure and speed remaining the same, it cannot be obtained by the addition of a high-pressure cylinder, as is very often erroneously assumed. The saving in fuel by compounding and the addition of a condenser should be between 15 and 20 per cent.

33. Comparison of Mechanical Efficiencies. — The mechanical efficiency of the simple engine of equal power compared with the duplex should be a little higher. There should be no appreciable difference in the economical

performance of the two types of engines unless the sizes are such as to render the duplex very small engines, in which event the economical efficiency of the duplex engine will suffer a loss. The duplex engine should operate more smoothly than the simple because of the more even turning moment at the crank-shaft. The simple engine of equal power, and if run at the same speed as the duplex engine, to secure the same coefficient of unsteadiness of rotation, will require a wheel 1.6 times heavier than the duplex, and necessarily a heavier shaft and larger bearings; this operates to reduce the mechanical efficiency of the simple engine.

34. First Cost.—For engines of equal power, under the same steam pressure and piston speed, the duplex engine will cost about 1.4 times as much as the simple engine, while the foundations will cost about 1.6 times as much as the foundations for a simple engine. It is to be noted, however, in selecting engines with reference to cost per horsepower that the price will be found to vary on either side of a minimum horsepower, which for ordinary engines will be about 500 horsepower. Owing to this fact, it sometimes happens that a very large duplex engine may be found to cost less than a single engine of equal power.

HIGH-SPEED VERSUS SLOW-SPEED ENGINES.

35. Classification.—The line of demarcation between high-speed and slow-speed engines is not clearly defined when referred to the number of revolutions made per minute or per second, for there is another characteristic that, when considered in connection with the number of revolutions per minute, assigns them to the class in which they belong. This characteristic is the manner in which regulation is secured. Engines with a releasing-valve gear, usually of the four-valve type, and regulated by a pendulum governor are classed as slow-speed engines, barring a few exceptions, while positive automatic cut-off engines,

regulated by means of a shaft governor, are invariably classed as high-speed engines. Engines regulated by means of a throttling governor are seldom classed as either high-speed or slow-speed engines; in fact, they usually run at speeds midway between those of the two first-mentioned classes of machine. In considering the relative merits of high-speed and slow-speed engines, the speed refers to the revolutions per minute, and *not to the piston speed*, for, as a matter of fact, the piston speed of modern slow-speed engines often exceeds that of the high-speed engine.

36. Purpose and Advantages of High-Speed Engines.—It may safely be said that the advent of electricity as a medium for the transmission of energy caused the development of the quick-running engine, and it is in the electrical field that it still finds its largest market. It is not confined to the electrical industry, however, and has been applied to nearly every service, either direct-connected or by belting. Its principal merits are comparative low first cost and small space required; its principal objections are wastefulness of fuel and need of constant attention. These objections are not universal, however, as there are some high-speed engines on the market that are quite equal in economy and in all other respects to any of the slow-speed engines. There are also some high-speed engines of the enclosed crank-chamber type that, to a great extent, are self-lubricating and demand very little attention; they usually take the vertical form. The majority of high-speed engines are not very economical machines and must be carefully watched.

37. Comparison of Valves for High-Speed Engines. High-speed engines are almost invariably fitted with a balanced valve, which is frequently a piston valve. It is claimed that this type of valve, if used on a horizontal high-speed engine, will begin to leak about the time the engine is paid for and will not improve with age, notwithstanding the many devices used to adjust the fit of these valves in their liners or casings.

The piston valve applied to the vertical engine has given better results as regards less leakage and resultant economy, but even here it has not been altogether satisfactory, principally because the system of regulation imposes varying travel to the valve and unequal wear on the internal surface of the casing or liner. Other systems of balancing are by means of pressure or cover-plates; these require very careful design and workmanship, but if properly designed and fitted, they are much superior to the piston valve. The clearance with the latter form of valve is usually much less than with the piston valve, but the clearance is generally large in all of them, and it is due to this fact that the periods of compression can be lengthened and the engine be made to operate very smoothly.

The Corliss valve has been used to some extent on high-speed engines, but the result has not been altogether encouraging, and in several instances they have been absolute failures, which was probably due to the fact that if pressure is allowed to remain on Corliss valves sufficiently long to force out the film of oil that is between the valve and the seat, it takes considerable force to move them.

38. Valve Motions.—The valve motion is invariably derived from an eccentric of variable throw and angular advance or from an equivalent crank, so hung as to give a nearly constant lead. The peculiar valve gears of the high-grade slow-speed engine, by virtue of which the valves move but little and very slowly after they have closed the ports, are seldom or never attempted in the high-speed engine.

39. Effects of Large Clearance Volume.—The necessary simplicity and desirable low first cost of high-speed engines has resulted, on account of the types of valves, in large and comparatively long steam ports, which increase the clearance volume and clearance surface. As was previously pointed out, high-speed engines have a high speed as regards the number of revolutions per unit of time; consequently,

the stroke must be shortened, which results in a high percentage of clearance volume; the frequency with which this clearance volume or part of it is filled with fresh steam affects the economy of this class of engine to some extent.

40. Effect of High Speed on Regulation.—Aside from the question of economy, one of the leading characteristics of high-speed engines is the regulation. On account of the high rotative speed, the regulator or governor has a much greater opportunity to effect changes in the speed; that is, if the high-speed engine is running 300 revolutions per minute while the slow-speed engine is making 100 revolutions per minute, the high-speed engine may be said to have 600 opportunities to adjust the steam supply while the slow-speed machine has only 200; or, in a unit of time, which may be taken as 1 revolution of the slow-speed engine, the high-speed engine has had 3 opportunities to adjust its speed. Consequently, the regulation of the high-speed engine is much superior to that of slow-running engines, even though they be fitted with governors equally sensitive. As a matter of fact, the better types of high-speed engines, as at present constructed, leave nothing to be desired in the matter of regulation for any possible commercial service.

41. Prevention of Accidents.—The increased risk of wear and the liability to accident due to their rapid motion, and especially when accidents do occur, the seriousness of their nature must be considered in connection with the high-speed engine. The prevailing tendency among builders of this class of engine is to reduce the possibility of accident by selecting higher grades of material, providing liberal wearing surfaces, which are case-hardened or oil-tempered, and using safe and thoroughly tested constructions embraced by massive and well-distributed framings.

42. Lubrication.—High-speed engines require copious lubrication, and unless careful provision be made to collect

the excess, great wastes may result in this direction. This is provided against to a great extent by providing splashers, oil guards, drip pans, and in some designs completely enclosing the running parts in oil casings; in some systems provisions are made for draining and collecting all oil in a separate chamber, where it is carefully strained or filtered and automatically returned to the bearings. In this so-called **return system,** a liberal stream or several streams of oil are kept running upon the bearing surfaces.

43. Accessibility of High-Speed Engines.—From the compact, rigid nature of the design of high-speed engines, they are not as accessible as the slow-running machine, but it cannot be argued that they are particularly difficult of access.

44. Influence of High Speed on Weight of Flywheel. Owing to the velocity of the high-speed engine and to the fact that the energy of a flywheel increases as the square of the velocity of the center of inertia, the wheels for high-speed engines can be made very much lighter and still obtain the same degree of unsteadiness as in the slow-running engine. This relieves the bearings of much dead weight and allows the shaft to be made smaller and makes the velocity of its rubbing surfaces much less.

45. Comparison of Economic Performances.—One of the elements in high-speed engines that, no doubt, contributes much to the economy of the machine is the little time allowed for initial condensation of each charge of steam and for the changes in temperature preceding each charge; some of the single-acting very quick-running engines have met with not a little success, their designers attributing it to the fact above mentioned. It must be borne in mind that the very highest duties and efficiencies have been obtained from the slowest running engines, as pumping engines, and many engineers contend that speed is not of vital importance in securing high economy. There is,

however, little basis for comparison between the two engines, for slow-speed engines can also be denominated as high-grade engines, while high-speed engines may be classed as low-grade engines; and while there may be no appreciable difference in the economical performance of high-grade engines when run at varying speeds, the economy of a high-speed engine would fall away materially if run at a slow speed.

46. Savings Due to High Speed.—The high-speed engine has it strongest claim over the slow-speed engine in its adaptability to direct-connected work, whether the connection be to electric generators, the shaft of a mill, or any industrial work. There is at once a direct saving not only in the first cost of the engine, but in saving due to the omission of transmission machinery, as jack-shafts, belts, or gearing, bearings and their foundations, and the continuing expense resulting from their attendance, lubrication, and repair. High-speed engines, owing to their greater steam consumption, demand a 20-per cent. larger boiler plant, which is an item of first cost to be considered. While the circulars of high-speed engine builders announce their capacity and willingness to build this type of machine for large powers, they are seldom met with in actual practice; the common range of power is from 60 to 200 horsepower, but they are occasionally built in units as large as 800 to 1,000 horsepower.

47. High-Speed Compound Engines.—High-speed engines are as commonly built compound and triple-expansion as are slow-speed engines, and they are more frequently built compound non-condensing than are slow-speed engines. The compounds are arranged both cross and tandem. With high-speed engines it is quite important to have spare parts on hand.

48. Economy of Slow-Speed Engines.—The slow-speed engine, which at the present time is almost invariably

of the four-valve drop cut-off or releasing-valve gear type, is most commonly chosen for all large units where continuous operation is required. In localities where fuel is expensive, even though the steam plant be used as a relay in case of failing water-power, the slow-running economical engine will be found. This condition is generally one requiring very careful study to obtain maximum commercial efficiency. The slow-speed engine admits of many, though practical, complications to the end of securing extreme economy of steam and high mechanical efficiency. The valves are usually so placed as to reduce the clearance volume and clearance surface to a minimum; great care is exercised to free the cylinders of water; steam jacketing of heads and cylinders is common. The polishing of the internal faces of heads and pistons in order to reduce the activity of the metal in receiving and imparting heat to the working steam, and thus reducing initial condensation, is sometimes resorted to; elaborate valve gears to give theoretical steam distribution are possible with the slow-speed engine.

49. First Cost of Slow-Speed Engines.—The slow-speed engine is much the larger, heavier, and more expensive machine, and usually costs $1\frac{3}{16}$ times as much as the high-speed engine of equal power. The foundations are also more expensive, but the boiler capacity need not be so large. The relative complete cost of high-speed and slow-speed power plants is not far from $50 per horsepower for high-speed and $70 per horsepower for slow-speed plants, the engines being simple non-condensing. The economical performance, assuming the engines to be in fairly good condition, should be about 30 pounds of water per horsepower per hour for the high-speed and 24 pounds of water per horsepower per hour for the slow-speed, bearing in mind, however, that the slow-speed engine will for a long time maintain its economical performance, while the high-speed engine will generally lose in efficiency.

50. Direct-Connected Engines for Dynamos.—The prevailing practice in the generation of electricity by steam power is towards direct-connected units, and the cost of the electrical generator is an item constantly urging higher rotative speeds; for this reason the slow-speed engine, a few years ago, was by several builders forced past its practical limits of speed. The impracticability of the move, as made apparent by constant breakage and short life, was soon recognized, and moderate speeds of about 90 to 100 or 110 revolutions per minute were returned to. These speeds are now seldom exceeded, and somewhat slower speeds are advised when long life and immunity from accidents are desired. Reducing the rotative speed of direct-connected engines operates to increase the diameter of the armature or revolving field of the electrical generator, and this, in turn, increases the first cost of the unit; it is here that the high-speed engine obtains its grip in the solution of the problem; but after passing a very moderate power, say 200 to 300 horsepower, the item of economy becomes of such magnitude that the more economical and more expensive slow-running unit establishes its ultimate value to the engineer or purchaser.

51. Attention and Workmanship.—Where the high-speed engine has its disadvantages, corresponding advantages may be found in the slow-running engine, and vice versa; slow-speed engines besides being much more economical do not require the close attention demanded of high-speed engines. With reasonable attention the slow-speed engine will give fair warning in many instances of approaching danger that will be developed too quickly in the high-speed engine to control. The slow-speed engine is more accessible and more readily adjusted; the workmanship need not be so exacting for equal results, the rate of mechanical depreciation is less, and the life and service of the machine is greater. The regulation of the slow speed engine is not equal to that of the high-speed engine, and they require very heavy wheels for the same degree of unsteadiness.

SELECTION OF ENGINES.

INTRODUCTION.

52. The problem of selecting an engine for a specified service is one demanding all the skill, experience, and forethought of the constructing engineer, for on it rests one of the vital and constant items of expenditure of the industry of which it forms so important a part. The elements determining the selection of an engine may be briefly mentioned here as the influence of the kind of service, the location, first cost, cost of fuel delivered at the boilers, steam pressure available, the duration of service, the facilities for repairs, the kind of labor available, the existing conditions, if additions or renewals, whether the engine shall be condensing or non-condensing, and if one large engine shall be used or whether the power shall be divided into several smaller units; each of these conditions will receive separate consideration.

KIND OF SERVICE.

53. Self-Selecting Service.—Evidently the service to which an engine shall be put is the first determining element in the selection of an engine. In many cases the kind of service at once determines the general type of machine, as, for instance, for hoisting, pumping, blowing, locomotive, and rolling service. There are many lines of service, however, in which the type is not self-selecting. The service may be divided into *continuous running with uniform load*, *continuous running with variable load*, *continuous running with uniform but increasing load due to growth*, and *intermittent running*.

54. Continuous Running With Uniform Load.—When the condition of continuous running with uniform load pertains, the first question arising is the cost of fuel. If this item of expenditure is great, the evident conclusion

is that the steam should be worked at as high a ratio of expansion as practicable, since the condition existing most readily permits economical working of steam machinery. Not only should the steam be worked expansively, but it should be done in a high-grade engine. Whether this be high-speed or slow-speed depends on the particular work the machine is called on to do. If the engine is direct-connected to an electric generator of not over 200 horsepower demand and the steam pressure is 125 pounds, while water is available for condensing purposes, maximum efficiency would be obtained by the use of a compound condensing high-speed engine having separate steam and exhaust valves and a governor controlling the admission valve only. Where economy is a desirable item, a high-speed engine having one valve for steam distribution is not a favorite. If the power demanded is large, the slow-speed high-grade engine should be the choice and the cost of fuel should dictate in a great measure the steam pressure, the ratio of expansion, and the refinements in all directions to the end of reducing all expenditures. If no water be available for condensing purposes, the compound non-condensing engine will prove a good investment; but to give good success, the compound non-condensing engine should have at least 140 pounds of steam pressure, and care must be taken that it is not too large for its work. It is better, as far as economy is concerned, to have this type of engine small rather than large for its work.

55. The prevailing practice for terminal pressures is 19 pounds absolute pressure in the United States, while 25 pounds absolute pressure is the practice of some good English builders. If expansion is carried below the atmospheric pressure, the low-pressure cylinder will prove a drag on the engine.

56. Continuous Running With Variable Load.—For continuous running with a variable load, the compound non-condensing engine should be avoided. For this service the compound condensing engine is most suitable, as it

works over a wide range of expansion without materially affecting its economical efficiency. The simple condensing engine is well suited for the purpose. If condensing water is not available and if the cost of fuel or demand for power is not sufficient to warrant the use of a cooling tower for condensing water, then the slow speed non-condensing engine working with a steam pressure of 100 pounds should be the choice.

57. Continuous Running With Uniform But Increasing Load.—For continuous running with a uniform load, which is expected to be increased, however, through extension of the business, we turn naturally to the simple non-condensing engine of high or low grade, depending on the cost of fuel, and arrange to make it into a duplex engine, a condensing engine, or a compound condensing engine, if water is available, as demands are made for increase of power. It would be questionable economy to provide for converting this machine into a compound non-condensing engine on account of the high pressure required to successfully operate this type of machine. In the absence of condensing water, the increased power could be most easily and inexpensively provided by making the engine a duplex.

58. Intermittent Running.—For intermittent running there is much dispute among engineers as to the best type of engine. Here also the cost of fuel enters as a determining element of considerable weight. One of the most familiar examples of intermittent running is the hoisting engine. In the coal fields we find the simplest types of engines with no pretence at economy, while in the Northwestern copper-mining district we find the most elaborate triple-expansion hoisting engines working with 185 pounds steam pressure. Compound condensing hoisting engines are very common in the Northwestern iron-mining district and in the South African gold-mining industry. In both of these localities the cost of fuel is a heavy item; in South Africa it is not only expensive but poor in quality. Condensing water is also very scarce and the prevailing practice

is to make the engines compound condensing, using cooling towers to extract the heat from the water, and to use the same water continuously. The contention in respect to high-grade multiple-expansion engines for intermittent work is that if they are more economical in continuous service, the same or nearly the same comparative margin in their favor will result in intermittent work, and it must be conceded that there is reason in the contention.

59. The problem of choosing an engine for certain classes of intermittent work demands the study of local conditions, in which the cost of fuel is probably the most important determining condition. If the cost of fuel is high, say $5 per ton, and the power demanded is large, and the duration of work between stops is 2 minutes, the high-grade multiple-expansion engine should be a paying investment even though the extra expense of providing cooling towers for condensing water be added. If the power required is small, the high-grade engine is very seldom or never used, even though the cost of fuel be large. For reversing rolling-mill service, the simplest and strongest type of engine is used, the high-grade or multiple-expansion engine seldom being used.

60. There is another class of service that might properly be mentioned under the head of intermittent-running engines. This exists in industries that, from their nature, can operate only during a season or part of the year, such as the beet, cane sugar, certain classes of wood fiber, and other industries dependent on season and soil to produce their raw material. There is usually considerable refuse in these industries, having more or less value as a combustible, which, if it cannot be converted into a more valuable byproduct, is generally used as steam-generating fuel. The amount of refuse will quite often determine the type of engine to be used in running the plant. It is usually in excess (except in the beet-sugar industry, where the refuse is a marketable byproduct), and then the simplest, strongest, and cheapest constructed type of engine is chosen.

61. Relay engines can be classified to some extent as intermittent-running engines; but as their term of service is often of considerable duration, especially when they supplement water-power, with the further condition that many water-powers are decreasing in force year by year, the high-grade slow-running engine is generally chosen for this service for large powers and the better type of high-speed engine where smaller power or occasional assistance is required.

INFLUENCE OF LOCATION.

62. Fuel Cost.—The influence of location must be considered in conjunction with other influences, the principal one of these being the price to be paid for fuel; we mention this first because it is the largest and most important influence. It is evident that location is the all-determining influence of fuel cost, as an engine located in the coal-mining districts may be selected wholly with reference to low first cost, owing to the low fuel cost, while with an engine located in the iron- and copper-mining districts the first cost is a comparatively insignificant item if the power requirement is large, owing to the high price of fuel. This is not only true of the mining industries, but of all uses to which the steam engine can be put. In the New England States the cost of steam coal is about $3.50 per ton. Many of the installations are large, and here we find the best types of large economical steam engines. In many instances, the location is very remote from any source of supply or repair, which fact conduces to the selection of more simple machines, subject, however, to other conditions.

63. Cost of Transportation.—The cost of transportation of large engines working at a high ratio of expansion may influence the selection in favor of the smaller high-speed engine. A peculiar condition of transportation, due to location, is frequently met in the Western mountainous districts, where it is required to install an engine, usually for mining and metallurgical purposes, subject to the condition

that no piece shall weigh more than 500 pounds, the reason for this being that the parts must be transported on mule back through dangerous and difficult mountain passes. The simplest types of engines working without expansion and ingeniously divided into a number of parts determined by the size of the machine are usually chosen for these locations. Even here the item of fuel cost enters, which, with perhaps even the water supply, must also be transported in a similar manner as the engine parts; hence, high-grade engines working expansively have sometimes been chosen. It is almost needless to say that the first cost of such an engine is quite high, and that it taxes the skill of the designer and builder to the utmost to produce it.

64. Existing Conditions.—There are other conditions that follow as a natural result of location and which are not dependent on the cost of fuel or transportation, but which, to some extent, go far in determining the type of engine. Existing conditions at any given location may fix the type of machine, as steam pressure available, speed desired, and feasible method of coupling, and the availability of condensing feedwater. The demands of the particular service may require a special type of engine and frequently a machine of special and peculiar construction to meet the demands of peculiar local conditions, such as size and height of buildings.

INFLUENCE OF FIRST COST.

65. The item of first cost is probably the first to enter the deliberations of the constructing engineer and purchaser and throughout the determination stands out as a condition against which all other elements of the determination are weighed. Whether it be given first place or made second to the cost of fuel depends in a great measure on the amount of power demanded in proportion to the quantity of finished product. When the outlay for fuel is small compared with other running expenses, due to a small demand for power in the particular industry, low first cost may prevail; but, on

the other hand, if the power requirements are large, even though the price of fuel is moderate, the saving in fuel will soon overrun the interest and extra depreciation charges against the engine high in first cost, but economical in the use of steam. Where a large amount of power is required, even though the cost of fuel is low, if no other conditions enter the problem, such as remote location from any source of supply or facilities for repair or if the labor obtainable is untrustworthy or incompetent to work an efficient installation economically, it will generally be a wise investment to install the high-grade slow-running engine, taking advantage of water for condensation if possible and compounding if available steam pressures are not too low for good results.

INFLUENCE OF FUEL COST.

66. Introduction.—As was pointed out in Art. **65,** the selection of an engine is to a great extent determined either by the first cost or the cost of fuel. It must be understood that for special services and extraordinary locations and conditions, special engines must be designed to meet the condition, and to a great extent regardless of first cost or the cost of fuel. These are special cases and are almost self-solving.

67. Cheap Fuel.—When fuel is cheap and, as is sometimes the case, must be burned to dispose of it, an engine low in first cost will be the natural choice. This feature should be combined with simplicity, and the engine should be of such design as to require as little attention as possible, since where cost of fuel is of little or no consequence, the whole steam plant is liable to be neglected or left to care for itself, particularly the engine. In this case, one of the single-acting vertical enclosed-crank type of engine or some similar simple engine would be the natural selection.

68. Dear Fuel.—When the cost of fuel is high, recourse must usually be taken to every known means for saving fuel

The extent of elaboration in that direction depends on the price of fuel. The various means and devices used at the engine to secure small consumption of fuel may be briefly mentioned, as high steam pressure and multiple expansion, steam containing 70° superheat, and steam-jacketing. It may be mentioned here that superheated steam and steam-jacketing are agents working in the same direction—namely, the reduction of initial cylinder condensation, and where one is used the other is superfluous; thus, to steam-jacket a high-pressure cylinder receiving steam of, say, 50° F., superheat would be a non-paying investment. A means of securing high economy in compound-engine performance is to provide an efficient reheating receiver.

Advocates of reheating receivers claim as high as 10 per cent. gain by their use. Low-pressure cylinders are frequently steam-jacketed, the pressure in the jackets being reduced to about one-half the boiler pressure. Many builders will not use jackets on low-pressure cylinders; the best practice, however, favors their use. Serrating or corrugating the outside of cylinder liners and jacket spaces of the heads to make them more active is sometimes practiced, as is also the polishing bright of surfaces exposed to the incoming steam. A thorough system of circulation and drainage of all jackets, as well as means for freeing them of accumulated air, are essential to high economy. Small units should be avoided, if possible, combining them into as few large units as practicable, since large engines are generally more economical than small engines. All cylinders, pipes, reheaters, etc. should be covered with a non-conducting covering.

69. If condensing water is available, it should be used and a vacuum of not less than 3 inches below the indication of the barometer should be obtained. If condensing water is not available, a cooling tower may be used, remembering that it is not possible to obtain a vacuum with cooling towers much better than 6 inches below the indication of the barometer. A primary heater may be used between the low-pressure cylinder and the condenser through which the feedwater

is pumped, thus extracting as much heat as possible from the steam. A steam separator should be used at the throttle valves, returning entrained water to the boiler. All other drains should be trapped to an automatic receiver pump to be returned to the boilers. Superheated steam, if heated to an effective stage, say 70° F. superheat, requires poppet valves on the high-pressure cylinder on account of the difficulty experienced in oiling and keeping any kind of slide valves tight. This type of valve is somewhat expensive compared with other types of valves. The superheat is so reduced by the time the steam reaches the low-pressure cylinder of a compound engine that Corliss or gridiron valves may be used on this cylinder. The degree of refinement to which it will be expedient for the engineer to go must be determined by the cost of fuel and the amount that the steam engine demands as against the interest and cost of repairs due to this extra outlay to secure the smaller outlay for fuel.

70. The growing practice in steam engineering is along the lines of greater economy of fuel, and the compound engine is fast finding its way in the coal-mining districts, even where the fuel is culm, which is delivered at a cost not much exceeding 50 cents per ton. Assuming that engineers in coal-mining districts have found it a paying investment to introduce high-grade engines for their purposes, there should be little question as to the advisability of elaboration in districts where the cost of fuel is not uncommonly eight times higher, provided no other militating influences present themselves.

INFLUENCE OF STEAM PRESSURE.

71. The steam pressure available may be a controlling factor where an engine is to be selected to replace an old or overloaded one. The steam pressure available is generally the element that determines whether an engine shall be a single-cylinder or multiple-expansion. Generally the steam pressure should be 100 pounds gauge pressure for effective

compounding with a condenser, while 135 pounds should be available for compound non-condensing engines and 160 pounds for triple-expansion condensing engines. It is a matter of fact that the compound condensing engine has such an economical range over wide variations of steam pressure that it can be proportioned to secure results approaching so nearly the triple-expansion engine that the additional outlay for a triple-expansion engine is questionable, except for special and otherwise favorable service, such as high-duty municipal pumping engines, where three cylinders, each actuating a separate plunger, conduce not only to extreme economy of steam, but to a steady flow of delivery water, thus avoiding shocks on both pumping machinery and delivery mains.

INFLUENCE OF DURATION OF SERVICE.

72. By duration of service is often meant the life of the engine for useful work. It frequently happens that machinery is installed for the purpose of developing a doubtful industry or on speculation, when the measure of the doubt will be the controlling influence in determining the degree of efficiency and first cost of the engine. Engines and machinery are sometimes sent to distant localities or those difficult of access to perform a service, and the expenses of transportation are such as not to warrant their return, for it must be borne in mind that when an engine has been used sufficiently to be called second hand, its selling price is reduced below its former value. In such cases, engines of low first cost, but strong, simple, and well built, are usually selected. It is manifestly important in such cases that accident and costly delay by breakage be provided against.

INFLUENCE OF FACILITIES FOR REPAIRS.

73. While this influence is usually not a strong or active one, it nevertheless must be considered and provided for either in the selection of the engine or provisions for keeping

it in good working order. This latter may be accomplished by providing the necessary skilled labor, tools, and supplies or by providing spare parts to replace such pieces as have been found by experience most likely to break.

Specifications for engines for foreign shipment frequently include the following parts: 1 pair of connecting-rod brasses for each end of rod; 1 crosshead shoe; 1 piston and rod complete; 1 complete set main and outboard bearing boxes; 1 eccentric strap; 1 complete releasing gear with dashpot, if of the releasing-gear type of engine; 1 steam valve and 1 exhaust valve for each engine or each side of a compound engine. If a condensing engine, the following additional parts are usually specified: 1 air-pump bucket and rod; 1 air-pump delivery deck with valve and guards complete; 1 set of India-rubber valves; duplicate sets of metallic packing to be furnished and all stuffingboxes designed for the use of fibrous packing and suitable glands to be provided.

74. While the influence of facilities for repairs may in many instances of small and even moderate-size plants dictate the simplest and strongest types of steam engines, it does not, in fact, obtain in the case of large installations, where the cost of fuel is high or even moderate. The difficulty can be met by providing either spare parts, facilities for repairs, or relay engines, and in many of the South African plants all three expedients have been found desirable. Even with the simplest and strongest types of engines, if the facilities for repair are not at hand, carrying spare parts is advisable, as accidents and defects are liable with the most carefully constructed engines.

INFLUENCE OF KIND OF LABOR OBTAINABLE.

75. This influence is one that to a great extent is controllable, but the possibility of being compelled to trust an expensive steam plant in unskilled hands even for short periods, and possibly for long ones, may have some weight

in determining the type of engine. High-grade economical engines generally require superior intelligence to maintain them in that condition where the item of extra first cost may be assured as a constant and continuing profitable investment. The condition of the kind of labor obtainable may, when other conditions operate against the selection of high-grade engines, carry the choice to the simplest and strongest types of machine, but except in extreme cases it is not of sufficient weight itself to determine a selection.

INFLUENCE OF EXISTING CONDITIONS AND PROBABLE EXTENSIONS.

76. This influence has to a considerable extent been dealt with in former articles, but it will not be out of place to retrace the subject, as it is important, and a little forethought in this direction may save a considerable future outlay with less satisfactory results. Additional engine power may be obtained in many ways. One of the simplest and least expensive practices is the addition of a condenser where condensing water is available, the resulting increase of power ranging from 18 to 25 per cent., depending on the type of engine and degree of vacuum obtained. A good condenser should add from $11\frac{1}{2}$ to 12 pounds to the mean effective pressure. A jet condenser at average temperatures will require about 28 times as much injection or condensing water as there is steam to be condensed; that is, 28 pounds of water at 70° F. will be required to condense 1 pound of steam at 19 pounds absolute pressure. The duplex engine is an excellent and efficient means of increasing power, as is also compounding by adding a low-pressure cylinder if condensing is practicable, even at the expense of a cooling tower. Another means of increasing the power and securing economy where provisions for the duplex engine have not been made is by installing the low-pressure side of a compound by means of an entirely separate engine and running them disconnected. Such engines act with sufficient precision for all practical purposes.

CONDENSING OR NON-CONDENSING ENGINES.

77. The question of whether to run condensing usually depends for its answer on the natural supply of cooling water available, and frequently the supply of cooling water determines the location of the steam plant. This is particularly true of large installations, which, if in large cities, are invariably on the water front. There is a decided gain by the use of a condenser, not only in fuel, but in first cost, as a condensing engine may be made, on the average, 20 per cent. smaller and almost 20 per cent. cheaper than one not provided with a condenser. The cost of air pump and condenser, if directly connected, the air pump being driven by the main engine, should be about 10 per cent. of the cost of the engine for average sizes, and should require about 2 per cent. of the power of the engine to drive it. If an independent condenser is used, which for many reasons is the most desirable arrangement, it should cost about 15 per cent. of the cost of the engine, considering here slow-running high-grade engines. High-speed and inferior engines require condensers larger in proportion, owing to the larger amount of steam used. For large plants, where abundant natural water is not to be had, the cooling tower may be used to extract the heat from the injection water and to use it repeatedly. Sometimes a pond or large pans or tanks on the roof top are devised for the purpose of cooling injection water. Both of these plans of cooling water are slow and very large areas are required and, on account of atmospheric changes, are very uncertain.

78. A number of quite effective water-cooling devices are now being regularly manufactured in units as large as 10,000 horsepower. The general principles of all are the same; they consist of a round or rectangular tower so devised that the delivery water, which the air pump delivers to the top of the tower, in its descent is divided into the greatest possible number of sprays or films; an artificial current of air traverses the surface of the water, extracting the heat from it and rendering it sufficiently cool for service as injection water. These towers, as now constructed, do not require much floor

space, a 300-horsepower tower occupying a space of 8 feet × 12 feet. Self-cooling condensers have also been used to avoid steam plants becoming a nuisance in thickly populated city districts, by virtue of the suppression of all exhaust noises, although the law allows an industry to make as much noise as is reasonable and unavoidable in the pursuance of its processes. The cost of fuel usually dictates the policy in regard to cooling towers. If a natural water supply is available, it should be taken advantage of for economical reasons. The jet condenser is the favorite for all land purposes on account of its low first cost as compared with surface condensers.

ONE LARGE ENGINE OR SMALL UNITS.

79. This is a problem not only demanding large experience and most careful study of the details, but it also demands a broad and comprehensive investigation regarding future conditions. The conclusion to put in one large unit is too frequently jumped at. Generally the fewer units there are, the more economical will be the plant; but the condition of service goes far to determine the selection in this respect. In many industries departments require the use of power only a short time or at intermittent periods during the day, and frequently demand widely varying speeds for best effect; they also frequently require to be operated overtime or all night. Long-distance transmission is also involved. In such cases it will often be found that subdivided power will give the best results and, as a matter of fact, some large industries in the New England States formerly driven by one large single unit have adopted the scheme of subdivided power, dividing a single 1,400 horsepower unit into 40 smaller units of varying power. The high efficiency of electricity as a power-transmitting medium has done much to solve the problem of transmission to remote and difficult points and has also contributed to the existence of large single units; but these units should not be so large that they will be run underloaded, for an underloaded engine is about as poor an investment as can well be imagined.

ENGINE FOUNDATIONS.

80. Purpose of Engine Foundations.—One of the most important items in the installation of engines is to provide a suitable foundation, not only in order to rigidly support the machine, but also to absorb the jars and shocks due to its reciprocating motion, because if these are not absorbed, it will result in injury to the engine in question and also to adjacent property, such as other foundations, walls, and structures of any kind resting on the adjacent soil.

81. Supporting Power of Soils.—The foundation, besides having sufficient mass to absorb vibrations, should be spread out over sufficient area to prevent settling. Accepted figures for the supporting power of various soils range from 1 ton per square foot for soft clay to 5 tons per square foot for compact sand bottom, while 200 tons per square foot is given as the supporting power of hard rock in thick strata.

82. Depth of Foundations.—The depth of a foundation will vary with conditions; it should go out far enough below the surface to be free from the effects of frost or the influence of loads borne by adjacent grounds. It is rarely less than 4½ feet for small engines and rarely exceeds 22 feet for the largest engines. A horizontal slow-speed engine foundation for a 40-horsepower engine should be 7 feet deep; a 200-horsepower engine foundation should be 9 feet 6 inches deep; a 600-horsepower engine foundation should be 11 feet 6 inches deep.

Different types of engines require somewhat different designs for their foundations; experience has been and is the only teacher in this subject.

83. Absorption of Vibrations.—High-speed engines stop and start the reciprocating parts many times a minute, and hence set up severe vibrations, which must be absorbed by the foundations. In many engines of this type very careful counterbalancing is used to balance the reciprocating

parts in the horizontal direction; this, however, leaves the counterbalance unbalanced in the vertical direction. This balancing in horizontal engines tends to prevent the engine sliding lengthwise upon its foundation, while in the vertical engine the revolving counterbalance tends to slide the engine upon its foundation in a horizontal plane, and the foundation in either case must be of sufficient mass to absorb the vertical and horizontal forces, since engine framings and subbases rarely have sufficient mass to absorb vibrations. When engines are on the upper floors of a building, the scheme of suspending a very heavy mass underneath the floor, but rigidly bolted through to the engine base, and thus making virtually a foundation suspended in air, has proven effective in preventing all vibrations. Care must be exercised in placing engines upon solid rock that some elastic medium, as layers of wood and hair felt, is used between the machine and the rock to prevent vibration of the engine being transmitted to adjacent property.

84. Foundation for Vertical Cross-Compound Engine.—In a vertical two-crank high-speed engine having the cranks placed opposite each other, the vertical forces act to vibrate the machine in a vertical plane parallel to the crank-shaft. Such foundations should be designed with footings, as *a* and *b*, Fig. 1, relieving the center of the foundation and thus preventing any tendency to rock on a central bearing. Fig. 1 gives the general foundation dimensions for a 700-horsepower vertical cross-compound engine running at 100 revolutions per minute.

85. Comparison of Foundations for Vertical and Horizontal Engines.—Horizontal engines usually occupy so much space in a horizontal plane that the supporting power of the soil will be very much above the load if the foundation is made as small as possible—that is, if the bolt holes through the capstones are 6 inches from the center of hole to the edge of the stone for a 1-inch bolt and 10 inches for a 3-inch bolt. There should be 4 to 8 inches of masonry outside the capstone, and if the sides are carried down

straight, sufficient bearing area will be covered, except in cases of alluvial soil. Vertical engines, owing to the small

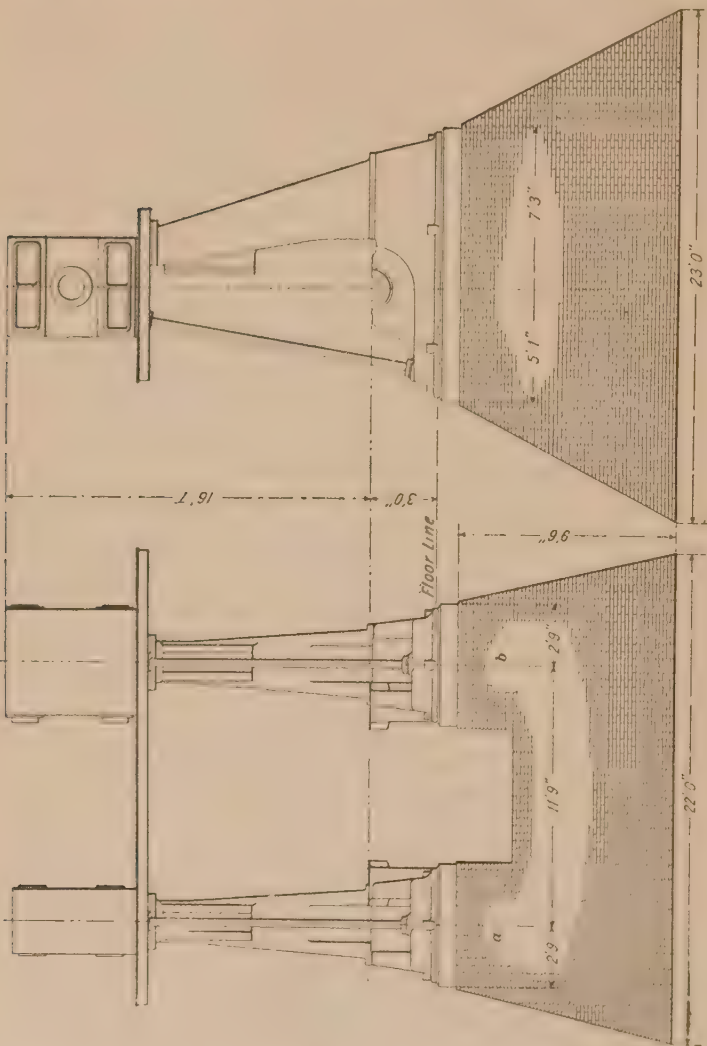


Fig. 1.

horizontal space required, should have deeper foundations than horizontal engines, and to secure sufficient bearing area.

the sides may be battered to any desired extent. Bearing surface for vertical engines should be carefully calculated with reference to the nature of the supporting soil, including, of course, the weight of the foundation itself as well as the engine that it supports.

86. Foundation Material.—The material of which a foundation is made depends very much on the location and the kind of material available. Brick is the most common material; dressed stone laid in cement mortar is sometimes used; concrete is growing in favor for engine foundations. When brick is used, it should be first quality hard brick laid in Portland cement mortar; lime mortars are not suitable for engine foundations on account of their tendency to disintegrate under vibration. Stone foundations should also be laid in cement mortar.

Foundations of concrete are coming more into use; they are constructed by first providing a level and suitable footing upon which a casing of timber, embracing the outlines of the foundation, is built. This is open at the top and bottom. The foundation bolts are suspended in pipes, old boiler tubes, or wood launders, leaving a space of at least 1 inch all around the bolt; successive layers of cement concrete are thrown in and well rammed until the desired height is reached.

87. Foundation Footings.—Foundation footings are in some cases required, most frequently on the water front, where it is necessary to go many feet deep to find a sufficiently hard bottom to support the load. This is accomplished most commonly by piling, which consists of driving long sticks or timbers, as *a*, *a*, Fig. 2, down to hard bottom, placing them $2\frac{1}{2}$ to 4 feet apart from center to center. A timber grating *b* is fastened to the tops of the piles and a layer of concrete *d* is deposited. Planking, as *c*, *c*, is sometimes put on the framing, which distributes the pressure, but it is considered objectionable, as it prevents any connection between the superstructure and the concrete and

increases the liability of sliding. The space between the piles is frequently filled with rubble, clay, or concrete, and upon this footing the foundation proper is built. A properly driven pile, well supported against lateral flexure, may bear from one-eighth to one-tenth the crushing load, which

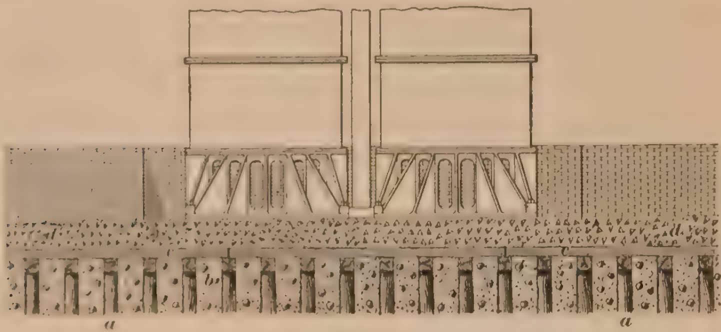


FIG. 2.

varies between 5,700 and 8,500 pounds per square inch. A pile 7 inches in diameter will bear about 12 tons. A pile can support a load of 25 tons when it refuses to move more than $\frac{3}{8}$ inch under thirty blows of a monkey weighing 1,200 pounds and falling 4 feet.

88. When rock is struck at a high level, special footings to prevent vibration must be constructed. An underlying stratum of timber or rubble or of both has been tried with questionable success; a layer of 2 or 3 feet of sand constrained laterally by a casing to prevent displacement has proved quite effective. The sand is also filled in around the sides of the foundation block. A heavy layer of asphalt is also effective in breaking engine vibrations before they reach the transmitting rock upon which the foundation is built.

89. Capstones.—Brick and dressed stone foundations usually require capstones to make a good job. These are usually granite and vary in thickness from 8 to 24 inches. Concrete foundations usually require no capstones. Instead of capstones, cast-iron sole plates are sometimes used. They

are usually thin, about $\frac{7}{8}$ or 1 inch thick, with an upturned ledge around the top to keep oil from the foundation and sufficient ribs below to give stiffness to the plate, and are provided with raised planed facings to match the engine parts. They are not more expensive than good capstones and are a superior job. Every precaution should be taken to keep oil from reaching the foundations, as it will dissolve the cement.

90. Many erecting engineers make a practice of setting the capstone for the outboard bearing from $\frac{1}{4}$ to $\frac{1}{2}$ inch lower than the actual figures called for and shim up the sole plate with wrought-iron strips or plates, the contention being that if the stone is set low the bearing can be shimmed up, but if a little too high it is a difficult matter to do anything with it but to chip off the top or take it up and reset it.

91. Foundation Bolts and Washers.—The bolts are always made of wrought iron, and should be of good quality, as Burden's best-best, Catasauqua, or some equivalent brand. Foundation bolts are usually made in length nearly the full depth of the foundation, and in important work they are made upset, that is, the threaded portion is made enough larger in diameter that the bottom of the thread is still a little larger than the body of the bolt. By this means the stretch due to the pull on the bolt is distributed over the long body and not only over the threads, as would be the case if the thread were cut on a rod of uniform diameter. Foundation bolts vary in diameter from $\frac{7}{8}$ inch in small engines to 4 inches in the largest types of land engines.

92. Foundation washers are commonly made of cast iron, but for small engines wrought-iron plates from $\frac{3}{8}$ to $\frac{3}{4}$ inch thick are used. In many locations it is not possible to provide pockets for access to the foundation washers; in such cases it is the best practice to provide cast-iron

washers *a*, Fig. 3. With this style of foundation washer it is possible to adjust the bolts to any desired height, or even to remove them and replace them at will. Some builders make a practice of tapping the top of the foundation bolt for an eyebolt to facilitate lifting or lowering the bolt in place.

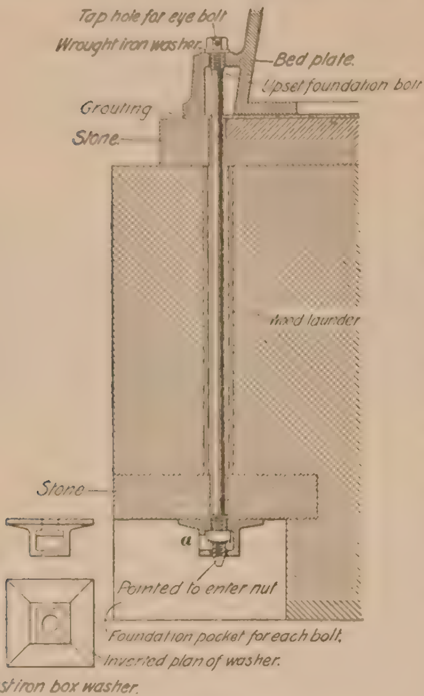


FIG. 3.

93. In large work it is quite important to have the foundation bolts removable, for if they extend through high framing or high bosses, it is necessary to lift the castings over the top of the bolts, which adds much to the cost of erection.

94. With many of the mining companies it is the custom to build foundations with pockets according to the drawing, but omitting the holes for the bolts entirely. When the capstones are leveled and grouted, they lay off the holes from the castings and drill holes for the bolts with a diamond drill. Sometimes the engine is erected, lined up, and grouted, and then the holes for the foundation bolts are drilled in place. With this arrangement it is essential in the design of the machine to see that holes for foundation bolts are not covered by any part or projection of the machine, or if this cannot be done, to provide a hole to pass the foundation bolt through.

95. Foundation Templets.—Foundation templets are used for the purpose of accurately locating the bolts, bringing

them to the proper height, and holding them in position while the masonry is being built. They are constructed of wood with blocks of varying height to suit the height of the engine bosses. The center line of the engine is carefully marked on the templet with correct relation to the bolts, and at right angles to it is marked the center line of the crank-shaft. Suitable marks and dowels to facilitate putting it together, if of such dimensions that it is necessary to ship it in sections, are also provided. The foundation bolts should not be allowed to hang on the templet, but rest on stone or bricks. The bolts, if they have no adjustment, should be set originally from 1 to 2 inches higher than required, as the gradually increasing weight of the foundation will sink the soil upon which it was started, and hence the bolts may not project through the engine casting unless this precaution be taken. Bolts when used in pockets or box washers should be pointed to facilitate entering the nut. Templets for out-board bearings or compound or triple-expansion engines are usually not connected, the relative locating being done from the foundation drawing. This work is usually done by the engine contractor, who sends only skilled men for this duty, but if not done by the engine contractor, it should be checked and approved by him at a sufficient time before the work of erection commences to have all defects, real or alleged, made good satisfactory to both parties.

96. Supporting the Templet.—The supporting of the templet must be left to the ingenuity of the erecting engineer; generally it should be supported outside of the foundation, but there is no real objection to building the supporting posts into the mass and sawing them off when the foundation has reached about 18 inches from the top.

97. Setting the Templet.—Setting the templet is a simple matter, but it must be carefully and exactly done, especially if the engine is to drive a shaft by belt or gearing. The first and most important thing to do is to have the templet exact; particularly the crank-shaft center line must be square with the center line of the engine. This can be

tested by measuring off from the intersection of the two lines 6 feet on the shaft center line and 8 feet on the engine center line and adjusting the lines with their intersection as an axis until the hypotenuse of the triangle measures exactly 10 feet. Then having given the templet the correct relative position and the correct levels, the only remaining thing to do is to set the center line of the crank-shaft parallel to the line shaft or its established line. This can be done by plumbing down from or to the line shaft and measuring at both extremities of the crank-shaft center line from that line to the plumb-lines. Some mechanics establish the center lines of the engine and the crank-shaft by stretching the lines considerably above the templet height and set the templet from these lines by plumbing down. It is not a very easy matter to stretch two lines in the air at exactly right angles, and hence it is a better plan to have the lines on the templet and exactly right, and then to work to the crank-shaft line as the most important one. Having properly set the templet, the bolts are passed down through the holes and the washers and nuts put into place. Each bolt must rest on a large stone slab. Old pipe or wooden boxes should be put around the bolts to allow considerable lateral adjustment of the bolts.

98. Placing the Engine.—When the foundation is built up to within 2 feet of the top, the templet is removed and the top of the foundation built and carefully leveled by means of sensitive levels and straightedges. If the engine is large and the wheel is in halves, one-half the wheel should be placed in the wheel pit first; then the framing, outboard bearing, and shaft may be placed and finally the cylinders and valve gears. The engine is leveled in a plane parallel with the center line of the shaft and cylinder by means of sensitive spirit levels and all parts resting on the foundation are wedged and shimmed up. The bolts are tightened down moderately and the space between the bed-plate and foundation, which will vary from $\frac{1}{4}$ to $\frac{3}{8}$ inch, is filled with grouting.

99. Grouting.—The grouting may be made of iron borings mixed with cement, sal ammoniac, sulphur, and water in about the following proportions: 2 parts of sal ammoniac, 1 part of sulphur, 5 parts of cement, and 40 parts of iron borings mixed with enough water to make a heavy paste. Sometimes melted sulphur alone is used, but one of the very best groutings and the most easily applied is pure Portland cement. The rust joint must be driven in, while the sulphur and cement will flow in, suitable dams being constructed to constrain it to its proper place. Bolt holes should be filled with liquid grouting. Some builders who use box bedplates fill the entire bedplate with concrete to give it solidity and to reduce the tendency to magnify knocks into pounds.

100. Setting the Outboard Bearing.—The setting and alining of the outboard bearing should be carried along with the progress of the other work, but its final adjustment is important and should be done last, by the aid of lines representing the center lines of the cylinder and crankshaft. All outboard bearings should be provided with a sole plate and means for lateral adjustment either by screws or wedges, while some are provided with means for vertical adjustment.

PUMPS.

(PART 1.)

GENERAL INTRODUCTION.

DEFINITION.

1. **Pumps** are machines for lifting or conveying fluids, and when not otherwise specified the word is generally understood to mean machines for lifting and conveying water.

WATER.

2. Water is a liquid composed of 1 part of oxygen and 2 parts of hydrogen. The weight of a cubic foot at its maximum density (39.2° F.) is 62.425 pounds; at 32° F., or the freezing point, water weighs 62.4 pounds per cubic foot, and at 212° F., or the boiling point, water weighs 59.7 pounds per cubic foot. Obviously pumping machinery can only handle water between the limits of the freezing and boiling points. Water is almost non-compressible; its compressibility is about .00005 of its volume under a pressure of 15 pounds per square inch, and it decreases with an increase of temperature; for practical purposes it may be considered as incompressible.

HOW WATER FLOWS INTO A PUMP.

3. Pumps are frequently so located that the water must flow into the pump cylinder by atmospheric pressure on the surface of the water external to the suction pipe; that is, by

the action of the pump a vacuum of more or less perfection is produced in the pump chamber. If the end of the suction pipe, which is the pipe connecting the pump chamber with the water, is submerged, the excess of pressure on the surface of the water outside of the suction pipe will cause the water to rise in the suction pipe until the pressure due to the weight of the column equals the pressure of the atmosphere.

THEORETICAL LIFT.

4. The pressure of the atmosphere is constantly changing. For practical purposes the pressure at sea level is taken as 30 inches of mercury, or 14.7 pounds pressure per square inch. Since a pressure of 1 pound per square inch is equal to that exerted by a column of water 2.309 feet high, the theoretical height that water can be raised by a perfect vacuum at sea level will be $14.7 \times 2.309 = 33.94$ feet. Since the atmospheric pressure becomes less as the altitude increases, it follows that the greater the altitude, the less the theoretical and practical lift by atmospheric pressure will be. To find the theoretical height in feet to which water can be lifted at any altitude, multiply the barometric reading in inches by 1.133.

5. For water holding foreign substances in suspension, or for other liquids, the theoretical lift can be found by dividing the theoretical height to which water can be lifted at the existing atmospheric pressure, as shown by the barometer, by the specific gravity of the liquid.

ACTUAL LIFT.

6. Since a perfect vacuum cannot be obtained on account of mechanical imperfections, air contained in the water, and the vapor of the water itself, the actual height to which it can be lifted is only about .82 of the theoretical height, which ratio is good only for pure water.

7. In the case of pumps located at the bottom of deep mines, the barometer will plainly show a greater pressure on

the bottom than at the surface, and hence a greater suction lift is possible at the bottom.

PUMPING HOT WATER.

8. Pumping hot water is a difficult problem and has positive limitations in the direction of lift and temperature. Whenever possible, the pump should be so arranged that the hot water will flow to it. The following table shows the theoretical possibilities at 30 inches barometric pressure:

INFLUENCE OF TEMPERATURE ON SUCTION LIFT.

Temperature. Degrees Fahrenheit.	Suction Lift. Feet.
100	28.0
150	21.0
170	17.0
190	10.0
210	1.5

In actual practice it is not possible to lift water at all whose temperature exceeds 180°. The reason hot water cannot be lifted is on account of the increased pressure of the vapor at the higher temperatures. Pumps required to handle hot water should be provided with suitable valves of vulcanized rubber for the lower temperatures and metal for the higher temperatures. Soft rubber valves are unsuited for handling hot water.

LIMIT OF HEIGHT TO FORCING.

9. Having considered the limit of lift by suction, the limit of height to which water or any other liquid can be forced will be discussed now. This height is *not* affected by

the atmospheric pressure and is only limited by the power available for forcing the liquid and the strength of the pump and the pipe connections.

GENERAL CONSIDERATIONS.

10. Before proceeding to give an account of some of the best and most modern types of pumps, let us consider for a moment what is required to be done in order that large volumes of water may be raised in the best possible manner and with the best possible economy. Among the first things that a practical engineer should know and among the last things he should forget is that in handling water within pipes he has a fluid which, while it is flexible to the greatest extent and is susceptible to the influence of power or force of greater or less intensity, and while it may be drawn from below and raised to a height above, and while it bends itself to the will of the engineer, will still refuse to do some things and which all the complicated appliances of the engineer have as yet failed to compel it to do. When enclosed within chambers and pipes to an extent that fills them, it will not permit the introduction of any more without bursting them. When enclosed within long lines of pipes, it will *not* suddenly start into motion or when in motion suddenly come to rest without shocks or strains more or less disastrous.

HISTORICAL.

11. Almost the first application of steam was to pumps used for raising water out of mines, and as these pumps had previously been entirely operated by horses, a basis of comparison was established by rating the power of the steam engine by the number of horses it displaced at these mine pumps. To enumerate even briefly the various machines for pumping water that have been developed in the past, many of which are famous, would be quite impossible for lack of space, and a description of their peculiar and prominent characteristics would be equally so, especially as they are only of historic interest. Conditions have so changed

as regards steam pressure, speed, and problems of manufacture and competition that out of the great mass of pump designs, some of which were excellent in many points, have been developed standard forms of pumps particularly adapted to each and every service.

CLASSIFICATION.

12. Pumps may be classified in a number of different ways, as according to their principle of operation, their general form, the power used to drive them, the methods of applying the power, the special class of work to which they are applied, etc. There being no universally accepted classification, no attempt will be made here to classify pumps, but the different forms of pumps described will be given the name by which they are most generally known.

STEAM PUMPS.

DEFINITION AND DIVISION.

13. **Steam pumps** are pumps in which the moving force is steam, which is applied to the movement of water without the intervention of belting or gear-wheels. Steam pumps are divided into two general classes known, respectively, as *direct-acting* and *flywheel-pattern* pumps.

DIRECT-ACTING STEAM PUMPS.

INTRODUCTION.

14. **General Description.**—The type of pump by far the most numerous is the **direct-acting pump,** by which is meant a steam-driven pump in which there are no revolving parts, such as shafts, cranks, and flywheels, or pumps in which the pressure of the steam in the steam cylinder is

transferred to the piston or plunger in the pump in a direct line and through the use of a continuous rod or connection. In pumps of this construction the moving parts have no weight greater than that required to produce sufficient strength in such parts for the work they are expected to perform; as there is, consequently, no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off steam in the cylinder during any part of the stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a steady, uniform pressure of steam throughout the entire stroke of the piston against a steady, uniform resistance of water pressure in the pump, the difference between the force exerted in the steam cylinders over the resistance in the pump governing the rate of speed at which the piston or plunger of the pump will move. The length of the stroke of the steam piston within these pumps is limited and controlled by the admission, release, and compression of the steam in the cylinder.

15. Development.—The direct-acting steam pump was invented by Henry R. Worthington in the year 1840 and was patented in 1841. A few years later Mr. Worthington developed and brought out what is now known as the *duplex* direct-acting steam pump. The objection to the single direct-acting pump was the fact that the action of the pump plunger or piston was an intermittent one; that is, the column of water was started into motion at the beginning of each stroke and came to a stand at the end of each stroke, thus not only making the flow of the water irregular, but also subjecting the pump and the connecting pipes and their joints to severe and often serious strains.

16. Duplex Pumps.—In the main, the construction of the steam and water ends of the duplex pump differs but slightly from that of the single direct-acting pump, but the mechanism that operates the steam valves is different and the effect on the water column is very different. The principle upon which the duplex pump operates is this: Two

pumps of similar construction are placed side by side; a lever attached to the piston rod of each pump connects to the slide valve of the opposite steam cylinder, and thus the movement of each steam piston, instead of operating its own steam valve, as in the single pump, operates the slide valve of the opposite cylinder. The effect of this arrangement is that shortly after the piston, or plunger, of one pump passes beyond the middle of its stroke, the plunger, or piston, of the other begins its movement, thus alternately taking up the load of the water column and producing a steady onward flow of water without the unusual strains induced by such a column of water when suddenly arrested or started.

17. Advantages of Direct-Acting Pumps.—The direct-acting machine is the simplest form of pump yet devised; its action most nearly harmonizes with the laws controlling the action of water, and in event of a conflict, the direct-acting pump will yield to the superior force of the water without serious resistance. The direct-acting pump being not only the simplest but most universally used type of pump it will be taken up first.

18. Disadvantages of Direct-Acting Pumps. — To obtain perfection of steam pumps it is necessary to use the steam so that by cutting it off within the steam cylinders and by subsequent expansion in the same or other cylinders, its expansive force will be developed to the highest limit and to the most economical extent. When that is done we have accomplished all that with our present knowledge of the steam engine can be done in the steam cylinders. It is in this respect, however, that the direct-acting steam pump of the ordinary type is anything but economical, its design requiring the carrying of the full steam pressure throughout the whole stroke. This drawback prohibits its use in places where a high economy in the use of steam is imperative. By the use of a so-called **high-duty attachment,** however, the ordinary direct-acting pump can be and is converted into a machine using steam expansively; a fair degree of economy is also obtained by compounding the steam end.

VALVE MOTIONS.

19. The Knowles Valve Motion.—Fig. 1 shows the steam end of the Knowles steam pump with the arrangement of the valve gear.

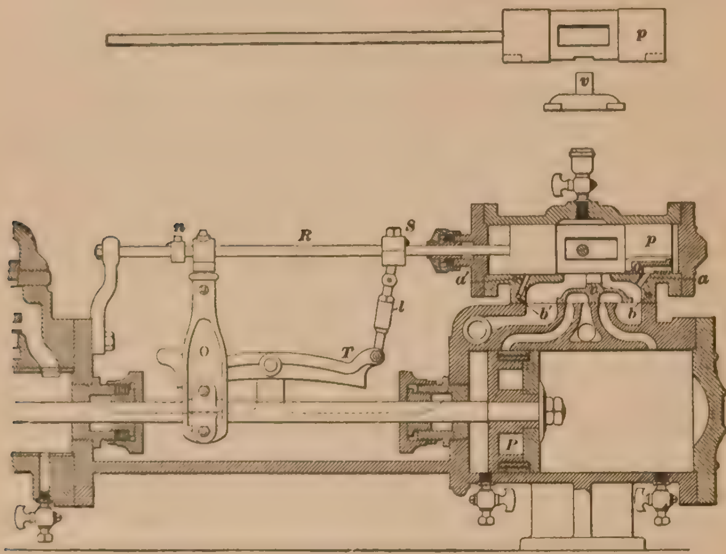


FIG. 1.

An auxiliary piston *p* works in the steam chest and drives the main valve *v*. This auxiliary, or **chest piston,** as it is called, is driven backwards and forwards by the pressure of the steam, carrying with it the main valve, which in turn gives steam to the steam piston *P* and operates the pump. The main valve *v* is a plain slide valve of the **B** form working on a flat seat. The chest piston has a rod *R* to which is clamped an arm *S*, the latter being connected to the rocker bar *T* by a link *l*. The main piston rod carries an arm *O*, which is provided with a stud, or bolt, on which there is a friction roller. This roller moves back and forth under the curved rocker bar with the motion of the main piston rod and lifts the ends of the bar, thus giving the chest piston a slight rotary motion just at the end of the stroke of the main piston.

Each end of the chest piston is provided with a port *o*, shown in the right-hand end by the partial section, and the solid part of the steam chest has four ports *a*, *b* and *a'*, *b'*, which open into the space in which the chest piston moves. The ports *a* and *a'* connect with the live steam space in the steam chest and serve as steam ports, while *b* and *b'* connect with the exhaust. In the position shown in the figure, the main piston has just reached that point of its stroke where the roller has acted on the rocker bar to rotate the chest piston. This has brought the port *o* (in the right-hand end of the chest piston) into communication with the live steam, admitting the latter to the space at the right of the chest piston. This steam drives the chest piston to the left and it carries the main valve *v* with it, thus exhausting the steam from the right of the main piston and admitting live steam to the left. When the main piston, under the action of this steam, approaches the right end of the cylinder, the roller lifts the right end of the rocker bar, thus rotating the chest piston so as to bring the port *o* in connection with the exhaust port *b* and the port in the opposite end of the chest piston in connection with the steam port *a'*. This drives the chest piston and main valve to the right, allows the steam at the left of the main piston to exhaust, and admits live steam to the right of the main piston again. The chest piston, as it approaches either end of its chamber, covers the exhaust port at that end, thus confining enough of the exhaust steam to form a **cushion** to prevent it from striking the end of the steam chest. The main piston also covers the exhaust port before reaching the end of its stroke, as shown in the figure, so that it is cushioned by the exhaust and prevented from striking the cylinder head. Special passages are provided for admitting the steam required to move the piston far enough to uncover the main ports on the return stroke. The arm *O* carries a collar that slides over the chest piston rod, and in case the steam pressure is not sufficient to move the chest piston, this collar will strike collars, as *n*, and thus move the valve. (One of these collars is just behind the arm *S*.)

20. The Cameron Valve Motion.—In the Cameron pump shown in Fig. 2, which possesses the advantage of

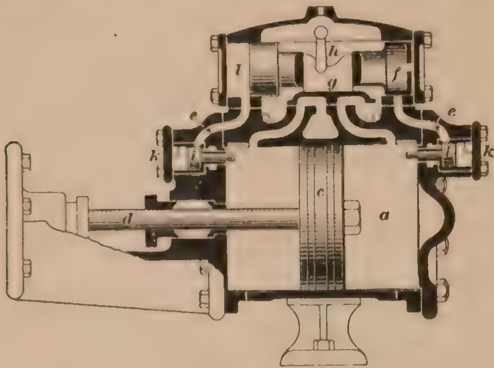


FIG. 2.

having no outside gearing, *a* is the steam cylinder, *c* the piston, *d* the piston rod, *l* the steam chest, *f* the chest piston, the right-hand end of which is shown in section, *g* the main slide valve, *h* the starting bar, connected with a handle on the outside, *i*, *i* the reversing valves, *k*, *k* the bonnets over the reversing-valve chambers, and *e*, *e* are exhaust ports leading from the ends of the steam chest direct to the main exhaust and are closed by the reversing valves *i*, *i*.

The action of this valve motion is as follows: The spaces at the ends of the chest piston *f* communicate with the live-steam space by means of small holes, one of which is shown in the right-hand section of *f*. By means of these holes, these spaces and the ports *e*, *e* leading from them are kept filled with live steam as long as the ports are covered by the piston valves *i*, *i*. In the position shown in the figure, the space in the main cylinder to the right of the piston *c* is in communication with the live-steam space in the steam chest; *c* is therefore moving to the left. When *c* strikes the stem attached to the valve *i*, it forces *i* to the left and uncovers the left-hand port *e*, thus allowing the steam at the left of *f* to pass out through the exhaust. The steam to the right of *f* then expands and drives *f* and with it the main

valve *g* to the left, thus reversing the action of the steam on *c*, which immediately begins to move back towards the right. Live steam is always acting on the piston *i*, so that as soon as *c* moves to the right, this steam pushes *i* back and covers the left port *e* again, after which live steam fills the port and the space connecting with it through the small hole in the end of *f*. When the piston *c* strikes the stem of the right-hand valve *i*, the main valve is again shifted to the right and *c* is started on its stroke to the left. Exhaust steam is confined in the ends of the cylinder to prevent the piston from striking the heads, in the same manner as in the Knowles steam pump.

21. The Gordon Steam Pump.—If the load is suddenly thrown off from the ordinary direct-acting steam pump

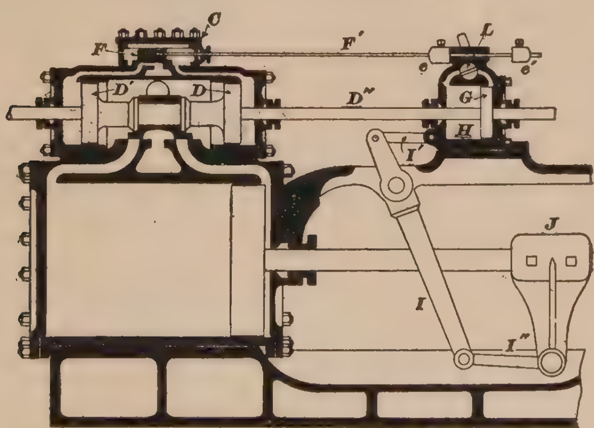


FIG. 3.

through any cause, as, for example, the bursting of the discharge pipe or the opening of a valve, so as to permit the water to discharge freely under low pressure, the steam is liable to drive the piston to the end of its stroke with so much force as to cause serious shocks or even to break some part of the pump. In order to overcome this danger the *Gordon* steam pump is provided with the arrangement shown in Fig. 3, which is called an **isochronal valve gear.**

In this gear the main valve is operated by a double chest piston $D\ D'$, which is actuated by steam controlled by an auxiliary slide valve F in the small steam chest C. F is provided with a valve stem F', to which two collars e, e' are fastened with setscrews. A slide H, which receives its motion from the main piston rod by means of links I' and I'', the lever I, and the crosshead J, strikes the collars e, e' near the ends of the main piston stroke, thus moving the auxiliary valve F and admitting steam to the chest piston $D\ D'$, which in its turn operates the main steam valve and reverses the motion of the main piston. In order to prevent the pump from running away when the load is thrown off suddenly, the slide H carries a cylinder in which works a piston G fastened to the rod D'' of the chest piston $D\ D'$. This cylinder, called the **cataract cylinder,** has a cock L that controls a passage joining its two ends, and by means of this cock the passage may be more or less closed, as desired.

The action is as follows: Assume the cataract cylinder to be empty; the piston G will then meet with no resistance and the machine will work as usual. At the end of the stroke the slide H will strike one of the stops e or e', thus shifting the auxiliary valve F and admitting steam to the piston valve $D\ D'$, which will move freely through its stroke and thus admit steam to the main piston for the return stroke. Now, if something happens to the water discharge, as, for example, the breaking of a pipe, the load will be removed from the pump and the main piston will be driven suddenly to the end of its stroke and thus be in danger of striking the head with enough force to break it. The object of the cataract cylinder is to overcome this danger. It is filled with liquid, which must be forced from one end to the other by the motion of the piston G. By partly closing the cock L a resistance is opposed to the passage of the liquid, and the motion of the piston G through its cylinder may be made as slow as desired; consequently, when the main piston moves too rapidly, the motion of the slide H will be transmitted to the piston G, which will shift the main valve

so as to shut off the supply of steam to the main piston and thus prevent the pump from running too fast.

22. The Marsh Steam Pump.—The **Marsh valve motion** shown in Fig. 4 operates without any connection to

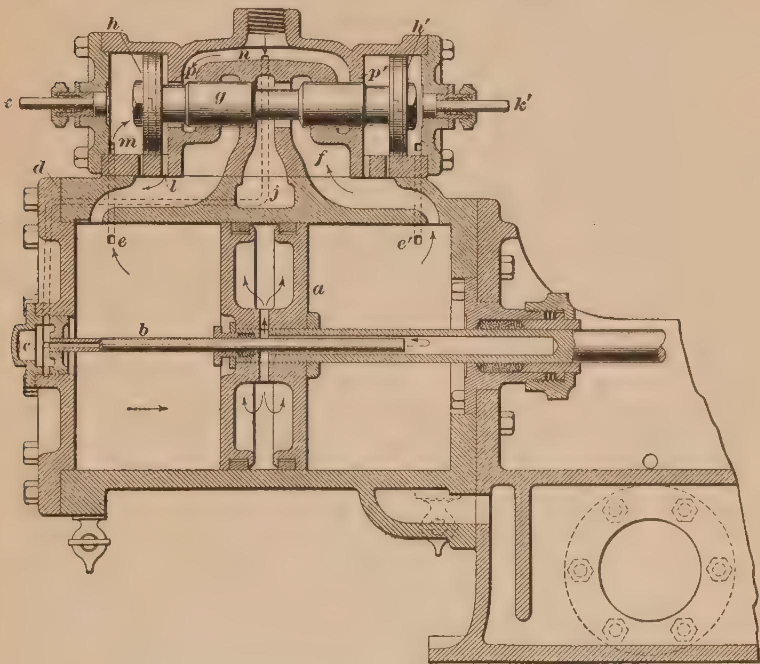


FIG. 4.

the piston or rod. The steam piston *a* is made in two parts, each section being provided with a packing ring so arranged as to provide an annular space between the two piston halves. Steam at boiler pressure is admitted within the pistons by means of the tube *b*, which is rigidly secured to and is in communication with the chamber *c*. A stuffingbox in the piston through which it plays prevents leakage into the main steam cylinder. A small port *d*, shown by dotted lines, supplies steam to the chamber *c*. When the piston is moving to the right, steam is entering from the space *n* through the annular opening between the reduced neck of the valve *g* and

bore of the left chest wall *p*. It is thus projected against the inside surface of the valve head *h* before escaping through the port and passing into the cylinder. Both the pressure and the impulse due to the velocity of the entering steam act on the valve head *h* and tend to force it to the left, thus tending to close the annular opening in the chest wall *p*. The steam flowing through the annular opening and port *l* into the cylinder also flows through the small ports *m* and *e* to the left of the valve head *h*. The steam entering through these ports is wiredrawn, so that its pressure is reduced, but it has a greater area of the valve head *h* exposed to its pressure than the steam on the right of *h*. Hence, the valve *g* moves to a position where the total forces acting on the two sides of *h* are equal and then remains stationary. The steam entering through the annular opening in the chest wall *p* is also wiredrawn, so that the pressure on the left of the piston *a* is below the full boiler pressure existing in *n*.

While the piston *a* is moving to the right, the steam on the right is exhausting through the port *f* into the exhaust port *j*. The exhaust is first closed by the piston running over the port *f*; as soon as this port is covered, the port *e'* leading to the right of the valve head *h'* communicates with the space within the piston containing steam at boiler pressure, and this live steam rushes into the space to the right of the valve head *h'*. Since the steam pressure on the left of *h* is less than the pressure on the right of *h'*, the valve moves to the left, and by doing so closes the left steam inlet, opens the left exhaust, and also opens the right steam inlet in the chest wall *p'*. The live steam admitted to the right of the piston *a* first brings it to rest and then reverses its motion. The tappets *k* and *k'* are used for moving the valve by hand in case the valve is stuck.

DUPLEX PUMPS.

23. Types.—Duplex pumps, like single direct-acting pumps, are made either as piston pumps or as plunger pumps. When made as plunger pumps, they may have either

inside-packed, center-packed, or outside-packed plungers. Piston pumps are preferred for moderate pressures, but for pumping against very high pressures the plunger pump is generally used.

24. Slide-Valve Worthington Duplex Pump.—Fig. 5 is a perspective view of a piston pattern Worthington

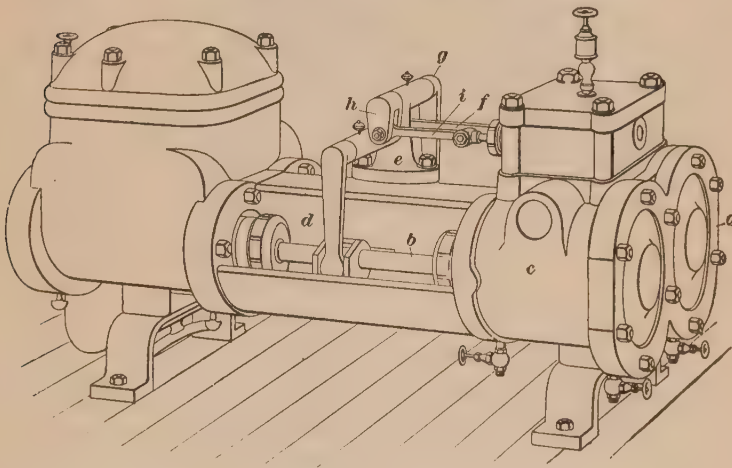


FIG. 5.

duplex pump, which shows the general arrangement of the valve motion. The two pistons always move in perfect harmony, and the steam valve for the cylinder *a* is worked from the crosshead of the piston rod *b* of the cylinder *c* through the lever *d*. This lever passes through the standard *e*; it is keyed to a shaft that carries a crank in line with it at the other side of the standard, and to this crank the valve rod *f* is attached. The valve rod in turn is hinged to the valve stem. The valve of the cylinder *c* is operated from the piston rod of the cylinder *a* through the lever *g*, the crank *h*, and the valve rod *i*

25. Fig. 6 is a sectional view through the center of the cylinder *a*, Fig. 5, and shows the construction of the steam end peculiar to the Worthington pump. Incidentally it also

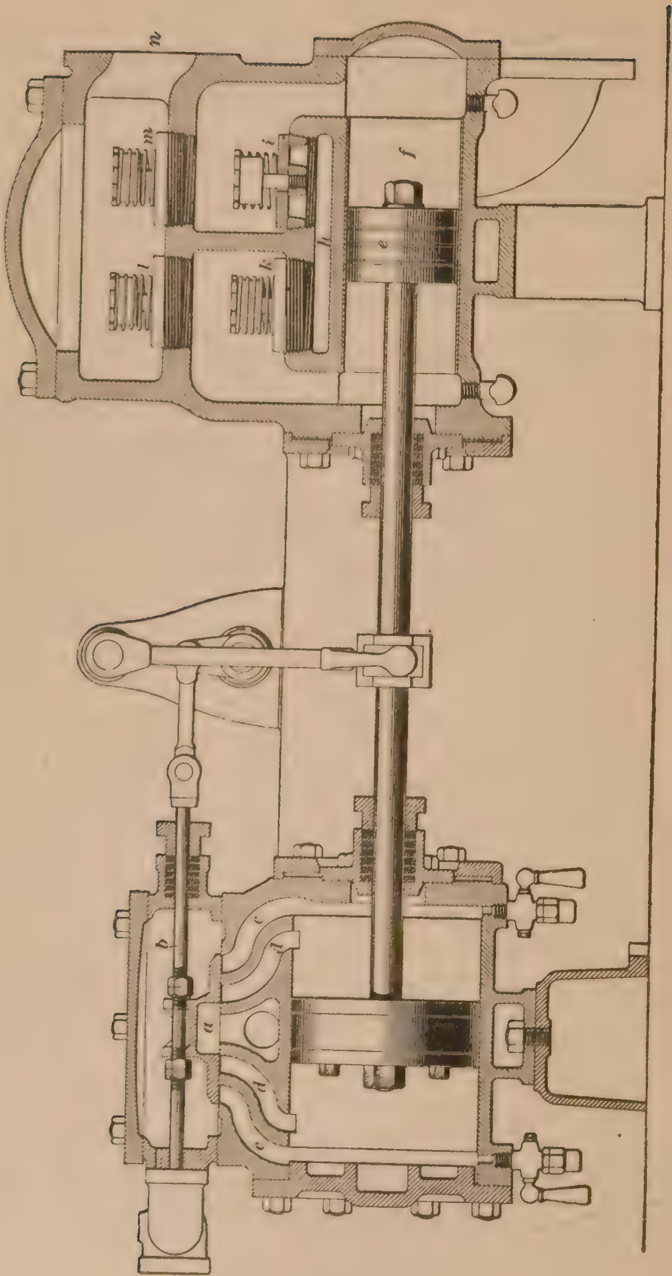


FIG. 6.

shows one form of construction of the water end of a double-acting pump. The steam valve *a* is a simple D slide valve operated by the valve stem *b*. There are two ports communicating with each end of the steam cylinder, of which the outer ones *c*, *c* are the steam ports and the inner ones *d*, *d* the exhaust ports. By this arrangement, when the piston approaches the end of its stroke, it covers the exhaust port and thus confines some steam in the cylinder that serves as a cushion. The valve has neither inside nor outside lap, and hence steam cannot be used expansively. The steam valve is carried along by coming in contact with check-nuts on the valve stem *b*, so placed that there is some lost motion between them and the valve. By this means the steam piston is caused to be at rest for a short time at the end of the stroke, which dwell allows the water valves to seat quietly.

26. In the water end the water is displaced by a piston *e* provided with suitable packing and working in the cylinder *f*. The water flows to the pump through the suction pipe connected to the lower nozzle and through the passage *h* to the suction valves *i* and *k*. The water is discharged through the discharge valves *l* and *m* into the discharge pipe connected at *n*. The operation is as follows: the piston *e* moving to the left, the suction valve *i* lifts and the discharge valve *m* remains closed, and water flows into the right-hand end of the cylinder. At the same time the water at the left end of the cylinder flows through the discharge valve *l*, which is lifted by the flow of water, while the suction valve *k* is kept closed. When the piston moves to the right, the suction valve *k* opens and the discharge valve *l* closes; at the same time the suction valve *i* closes and the discharge valve *m* opens. The pump is thus seen to discharge and take water during both strokes of the piston, and hence is double-acting.

27. Piston-Valve Worthington Duplex Pump.—Fig. 7 shows the steam end of a Worthington duplex pump

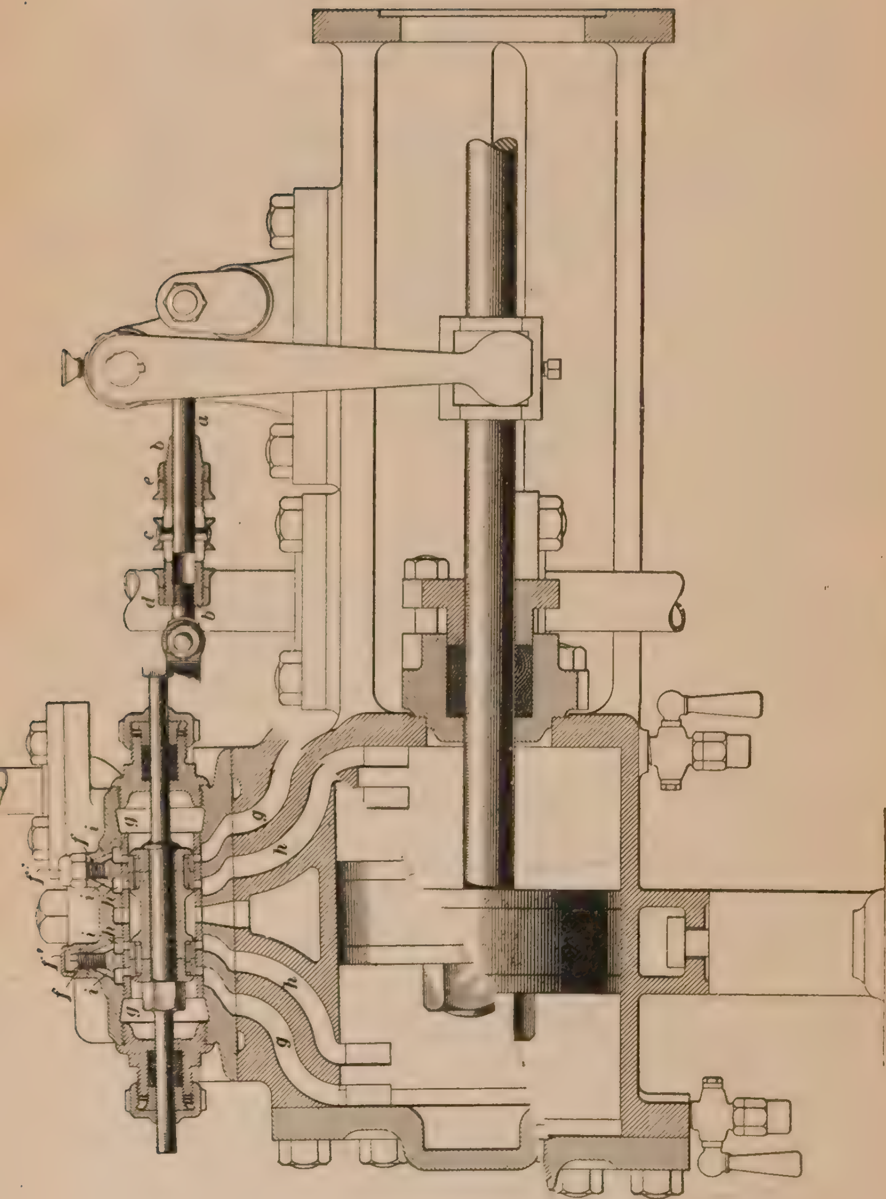


FIG. 7.

in which piston valves are used instead of slide valves. The valves are operated in practically the same manner as those of the pump shown in Fig. 5, but the lost motion instead of being between the valve and stem is obtained by a special construction of the valve rod *a*. This rod is divided into two parts. The part attached to the valve stem carries a slotted yoke *b*; the part attached to the crank is free to slide within the yoke and carries a collar *c* pinned to it. The collar *c* alternately strikes against the check-nuts *d* and *e* on the yoke *b* and then carries the valve with it. The lost motion is quite large, as the valve needs to be moved but a slight amount.

28. The length of stroke is adjusted by the use of the so-called **dash relief valves** *f*, *f*. These valves control passages *i*, *i* connecting the steam ports *g*, *g* and exhaust ports *h*, *h*, and are set by trial to the correct position and then locked with the cap nuts f', f'. The action is as follows: When the piston on its exhaust stroke covers the port *h*, no further exhaust can take place, and the steam will be compressed between the piston and the cylinder head. The location of the ports *h*, *h* is so chosen that the compression will stop the piston just short of the cylinder head at the highest speed at which the pump can operate. It is evident that when the pump is working at slow speed, the compression being the same as at high speed but the momentum of the moving parts being less, the piston will stop earlier than at high speed; i. e., the stroke is shortened. The dash relief valves prevent this shortening by providing an escape for the exhaust steam after the exhaust ports *h*, *h* are closed. It is thus seen that by them the amount of compression is regulated to suit the speed of the pump and the length of stroke is thus kept constant.

Dash relief valves are applied to pistons over 14 inches, as a general rule, and are used with slide-valve pumps as well as with piston-valve pumps. In either case they simply control a passage by which the exhaust port and steam port communicate.

MULTIPLE-EXPANSION DIRECT-ACTING STEAM PUMPS.

29. Purpose.—In the simple direct-acting steam pumps, no use can be made of the expansive force of the steam. They are, therefore, very extravagant in the use of steam, and in order to overcome this waste to a greater or less extent, many of the larger pumps are made with either compound or triple-expansion engines.

30. Compound Pump.—Fig. 8 shows a common method of arranging the cylinders for a **compound duplex** pumping

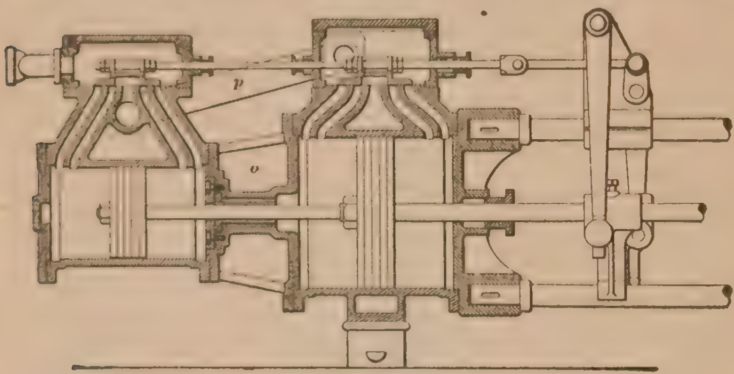


FIG. 8.

engine. The engine for each pump is made with two cylinders arranged tandem, the valves for both cylinders being driven from the same valve stem. The high-pressure cylinder is placed outside and connected to the low-pressure cylinder by a cast-iron yoke, or spacer *o*, which forms one head for each cylinder. The high-pressure piston rod passes through a sleeve in this spacer, as shown; this sleeve is held in its place by its flange being gripped between the spacer and a plate bolted on to the latter; otherwise the sleeve is free to adjust itself slightly, being a free fit in the body. The exhaust passes directly from the high-pressure cylinder through the pipe *p* to the steam chest of the low-pressure cylinder. Since there is no cut-off in either cylinder, the back pressure on the high-pressure piston is at all times equal to the pressure on the low-pressure piston, neglecting

the resistance to the flow of steam through the ports and the pipe *p*.

Since the volume of steam admitted during each stroke is equal to the volume of the high-pressure cylinder, and this steam, when exhausted, just fills the low-pressure cylinder, it is evident that the number of expansions is equal to the ratio of the volume of the low-pressure cylinder to that of the high-pressure cylinder. Also, since the length of stroke is the same for both cylinders, the number of expansions is equal to the ratio of the areas of the low- and the high-pressure piston. The usual number of expansions for small and medium sizes ranges from two to three. For large sizes four expansions are sometimes used.

31. Compound pumps are also made in which the cylinder arrangement is just the reverse from that shown in Fig. 8. In some of these compound pumps the high-pressure cylinder has no separate steam and exhaust ports; the compression and adjustment of length of stroke then takes place in the low-pressure cylinder.

32. Triple-Expansion Pump.—In triple-expansion pumping engines of the direct-acting class, the arrangement shown in Fig. 9 is sometimes adopted for the steam end. This design makes all the pistons accessible and at the same time avoids the use of a stuffingbox between the high-pressure cylinder *A* and intermediate cylinder *B*. The low-pressure piston and intermediate piston are connected by the piston rod *c*, and the low-pressure piston is connected to the high-pressure piston rod by the side rods *e*, *e* and the yoke *f*. The piston rod *c* is nicely finished and ground and works through a cast-iron bushing *g*, which is a nice fit. This bushing can move sidewise slightly so as to accommodate any want of alinement between the two cylinders. At the same time it prevents leakage of steam from the intermediate cylinder *B* to the low-pressure cylinder *C*. The low-pressure and high-pressure stuffingboxes are quite accessible. Access to the different pistons is had by removing the covers *h*, *i*, and *k*.

33. Fig. 10 is a vertical longitudinal section of the pump whose piston rod and cylinder arrangement is shown in Fig. 9 and shows the steam distribution in this form of pump. In the illustration, *A* is the high-pressure cylinder; *B* is the intermediate-pressure cylinder; *C* is the low-pressure cylinder; *d* is the high-pressure distributing valve; and *e*, *e* are the high-pressure cut-off valves. Steam enters through the center of the valve *d* and passes through the port *f* and through the cut-off port *g* into the high-pressure cylinder by way of the port *h*. The cut-off is effected by turning the rotary valves *e*, *e*. Exhaust from the high-pressure cylinder takes place through the ports *j*, *j* and thence into the high-pressure exhaust *k*, which leads to the inside of the intermediate steam valve *l*. The valve *l* is a rotary valve designed to distribute the steam exactly in the same manner as a common D slide valve. The intermediate- and low-pressure cylinders are not provided with cut-off valves. The exhaust steam from the intermediate cylinder passes out through the port *m* into the

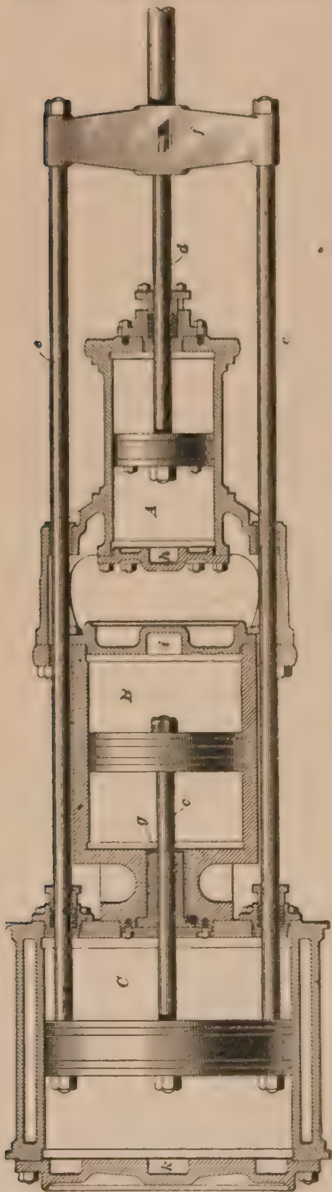


FIG. 9.

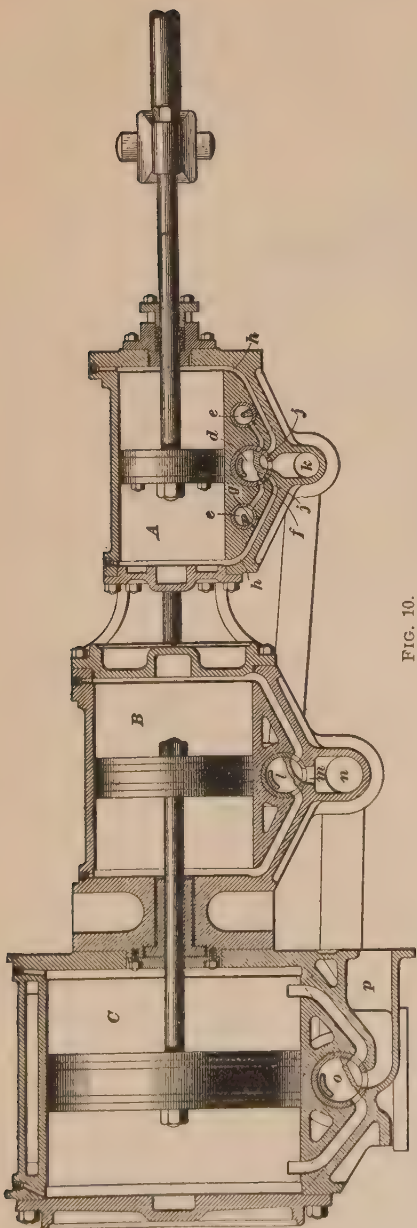


FIG. 10.

exhaust pipe *n*, and thence to the center of the low-pressure distributing valve *o*. From the low-pressure cylinder the steam is exhausted into the exhaust chest *p* and thence into the condenser or atmosphere. Dash relief valves, not shown in the illustration, are provided on the low-pressure cylinders only. The distributing valves are worked as usual from the pump on the opposite side, while the cut-off valves are worked from the pump on which they are placed.

34. Cross Exhaust. Compound duplex direct-acting pumps are occasionally provided with a so-called **cross-exhaust** connection, the purpose of which is the keeping of a more uniform pressure in the steam chests of the low-pressure cylinders than obtains otherwise. As shown in Fig. 11, it is simply a pipe *a* of ample size, which is provided with a valve *b* and connects the steam chests of the low-pressure

cylinders. The exhaust from the high-pressure cylinders *c*, *c* flows through the exhaust pipes *d*, *d* into the low-pressure steam chests *e*, *e*, but as the steam pressure there drops towards the end of the stroke, there is a diminishing of the

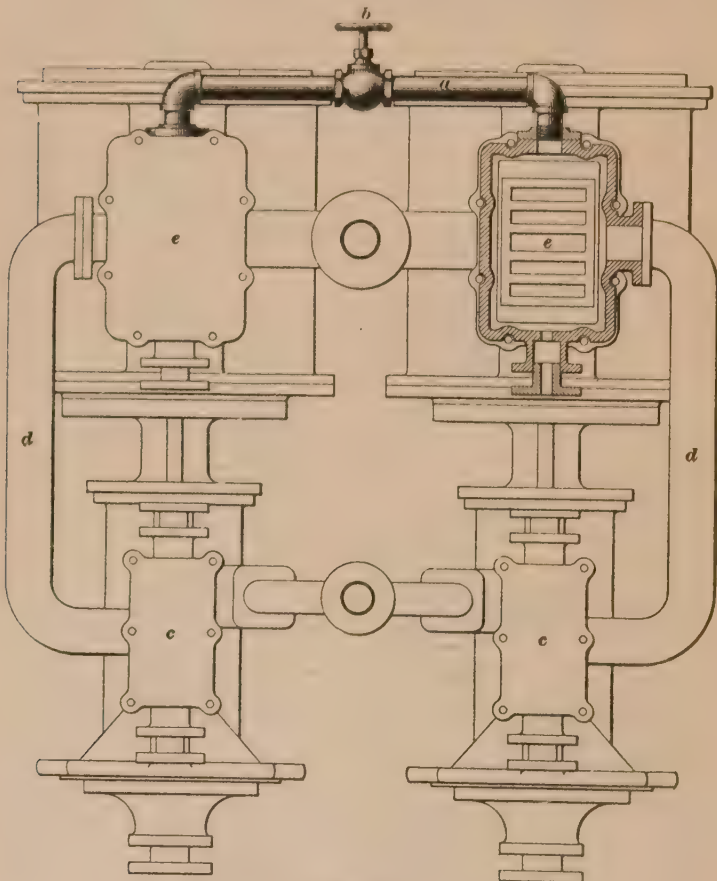


FIG. 11.

impelling force on the steam pistons of the low-pressure cylinders that tends to shorten the stroke. With the valve *b* open, the exhaust from the high-pressure cylinder of one pump can pass to the low-pressure steam chest of the other

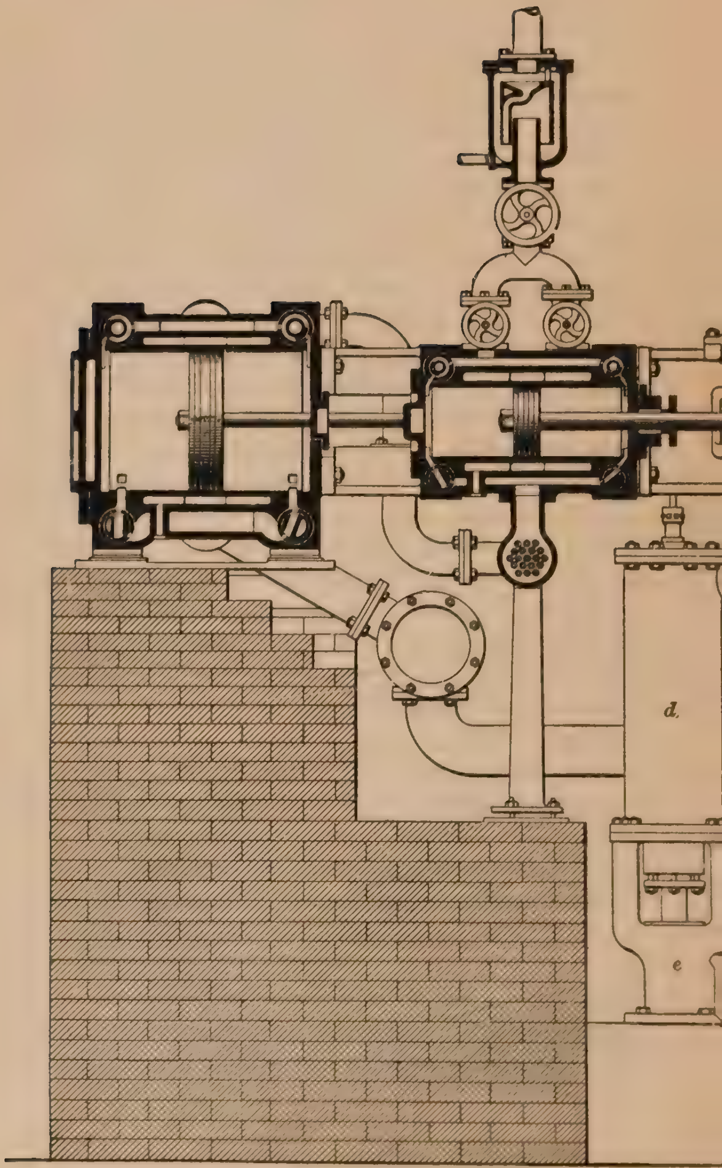
pump just when the pressure in that steam chest commences to drop, and in consequence the pressure will be kept more uniform, which results in a steady and uniform motion.

HIGH-DUTY ATTACHMENT.

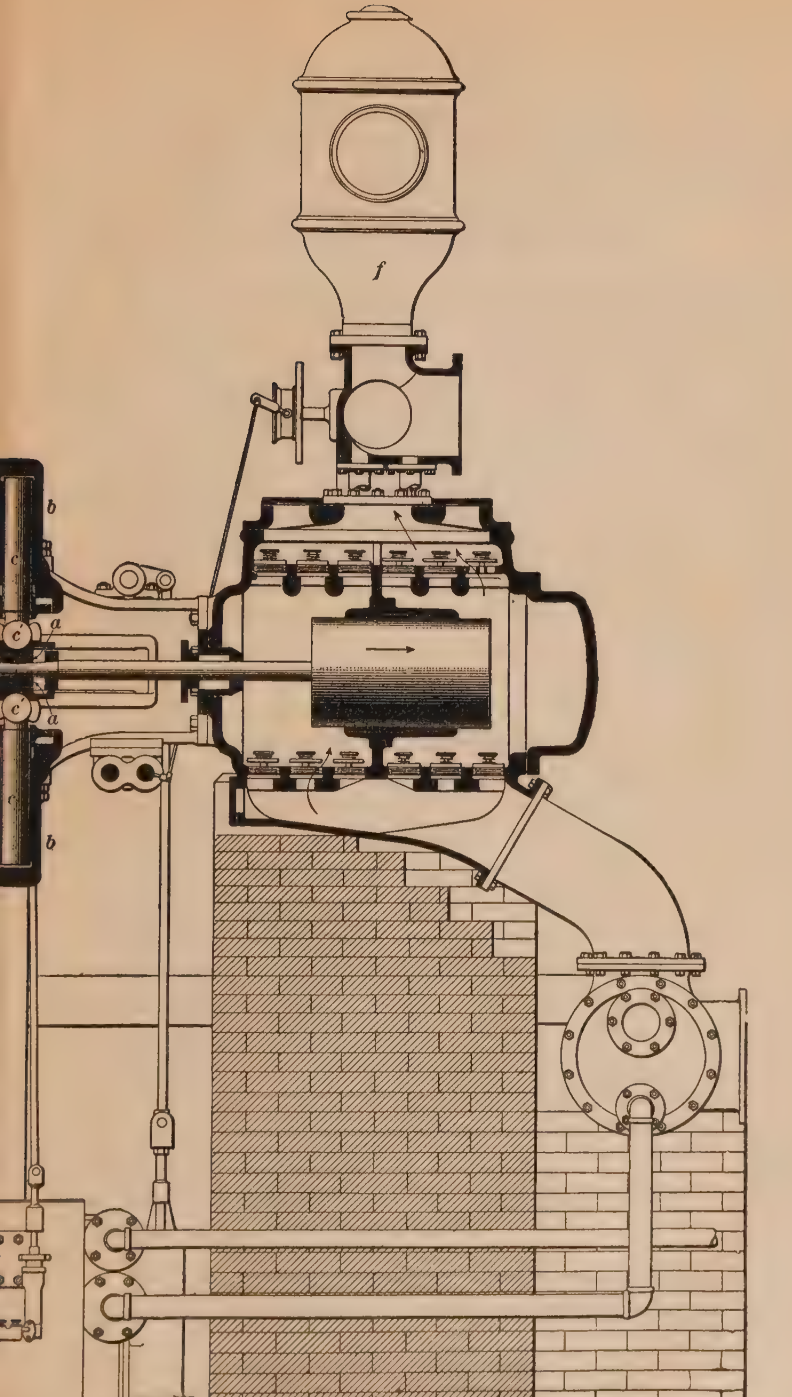
35. Purpose.—The direct-acting pump, as previously stated, is one of the simplest machines for pumping liquids, but in order to work at its best requires steam at full boiler pressure to be carried to nearly the end of the stroke. In consequence, if viewed from the standpoint of steam consumption, it is a very wasteful machine. The direct-acting pump is made more economical by making it compound or triple expansion, but even with these arrangements it is not possible to secure the high ratios of expansion which are necessary for extreme economy in the use of steam, and hence of fuel, and which are demanded in large pumping plants for commercial reasons. In the ordinary steam engine, and also in the flywheel pattern of pump, power is stored up in the flywheel at the beginning of the stroke and given out when expansion begins, in order to have a uniform turning of the engine shaft, or a nearly uniform force acting upon the water piston in case of a pump. In a direct-acting steam pump, however, there are no heavy moving parts similar to a flywheel, and hence ordinarily no uniform impelling force can act on the water piston if steam is cut off early in the stroke. This defect led to the design of the **high-duty attachment,** which is simply a device that stores up power during the first half of the stroke and gives it out again during the second half, thus allowing steam to be used expansively in the steam cylinders.

36. Construction.—The high-duty attachment in actual use was designed by Mr. J. D. Davies in 1879 and taken up and perfected by Henry R. Worthington. It is shown in Fig. 12 applied to a compound direct-acting pumping engine

fitted with Corliss valves and cutting off early in the high-pressure and low-pressure cylinders. The piston rods are arranged so as to avoid internal stuffingboxes, and, in consequence, the pistons are accessible without having to dismantle the pump. The two piston rods of the low-pressure piston and the high-pressure piston rod are attached to a common crosshead *a*, which runs in guides between the pump chambers and high-pressure cylinders. On this crosshead and opposite to each other are semicircular recesses. On the guide plates are cast two journal-boxes, one above and the other below the crosshead, equally distant from it and at the point equal to the half stroke of the crosshead. In these journal-boxes are hung two short cylinders *b*, *b* on trunnions that permit the cylinders to swing backwards and forwards in unison with the motion of the plunger crosshead. Within these swinging cylinders are plungers *c*, *c*, which pass through a stuffingbox on the end of the cylinders, and on their outer end have a rounded projection c', which fits in the semicircular recesses in the crosshead. Consequently, as the crosshead moves back and forth, it carries with it the two plungers *c*, *c*, which, in turn, tilt the cylinders backwards and forwards. These swinging cylinders are called **compensating cylinders**; they are filled with water or with whatever fluid the pump may be handling. The pressure on the plunger within the compensating cylinders is produced by connecting the compensating cylinders through their hollow trunnions with an **accumulator** *d*, the ram of which moves up and down as the plungers of the compensating cylinders move in and out. The accumulator used is of the differential type; that is, it has a small cylinder *e* filled with oil or water in which its ram moves, and above it has a much larger cylinder *d* filled with compressed air. On the top of the ram of the accumulator is an enlarged piston rod carrying a piston, which fits closely in the air cylinder. From this construction it follows that the pressure per square inch on the ram of the accumulator will be the pressure of the air in the air cylinder per square inch multiplied by the ratio between the area of the air piston and the ram



10B-291 § 34 T 6C § 38



. 12

of the accumulator. The ratio of these areas is made to suit the particular service for which the pump is constructed. The pressure in the air cylinder is controlled by the pressure in the main delivery pipe of the pump, as it is connected to the air chamber *f* on the main delivery pipe.

37. Operation.—The operation of the high-duty attachment will now be explained. Suppose the pump is about to begin the forward stroke. At this time the water cylinders will be turned so as to point towards the steam cylinders, with their plungers at the extreme point of their outward stroke and at an acute angle with the line of motion of the crosshead, and with the full pressure of the accumulator load pushing them against the advance of the crosshead. As the pump plunger begins its forward stroke, each forward movement it makes changes the angle of the compensating plungers, until at mid-stroke the two plungers will stand exactly opposite each other and be at right angles with the pump plungers, in which position they can neither retard nor advance the movement of the plunger. Now, as the pump plunger passes the mid-stroke position, the compensating plungers begin to push the pump plunger along, whereas before and up to mid-stroke they resisted the movement of the pump plunger. This force increases constantly, until at the extreme end of the forward stroke, and when the compensating plungers are, as at beginning, at their most acute angle, they exert their greatest force in helping to aid the pump plunger in its outward movement. The return stroke of the pump is made under precisely the same conditions as the forward stroke. It is readily seen that at the beginning of the stroke and up to mid-stroke, work is being done in pushing the compensating plungers inward, and that after the crosshead passes the mid-position, work is being done by the compensating plungers. The effect of this is a nearly uniform force on the pump piston with a varying pressure in the steam cylinders.

38. An important feature connected with the use of the compensating cylinders is that the results obtained by their

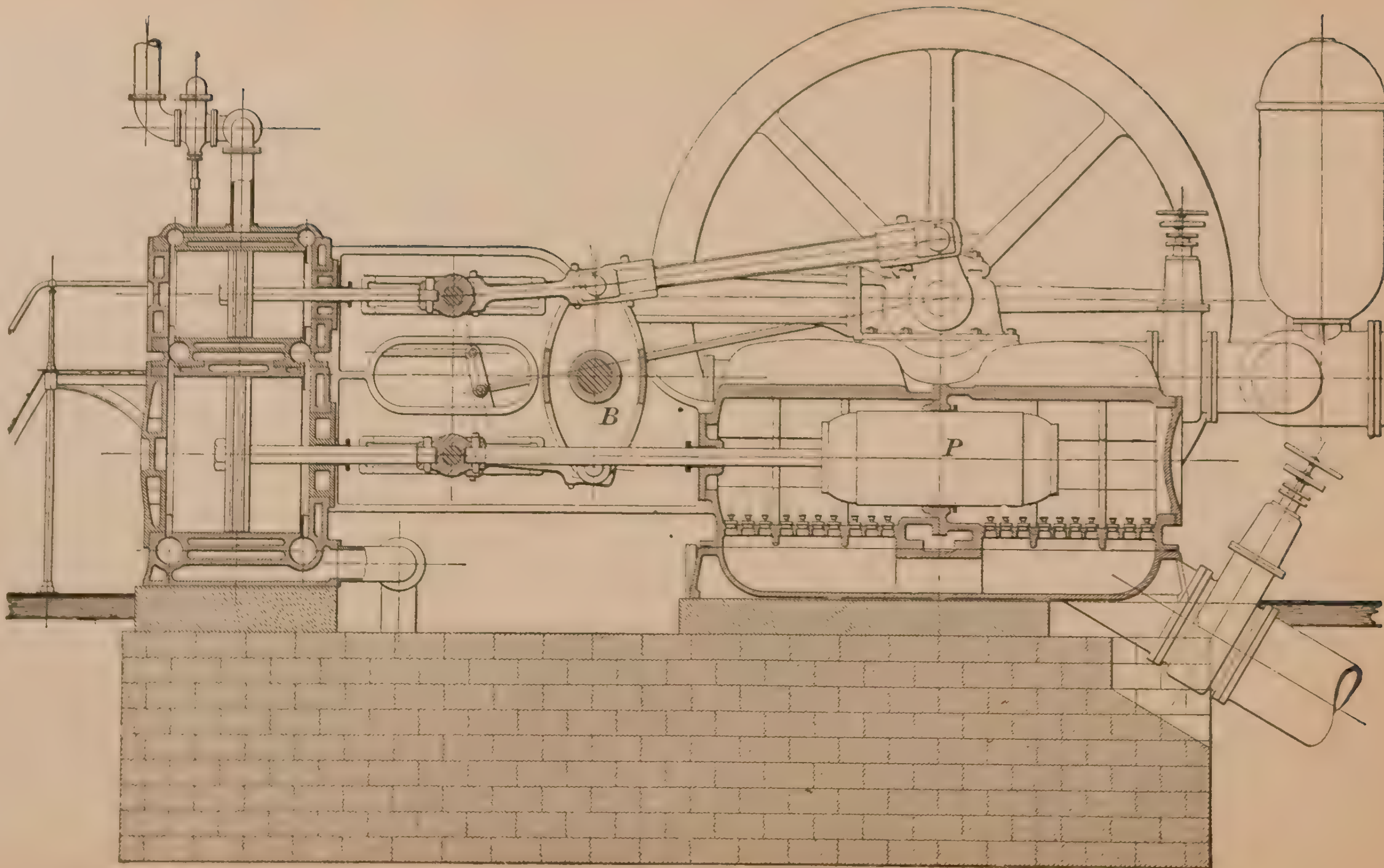


Fig. 13.

use are independent of the speed, in which respect their action is better than that of a flywheel. The high-duty attachment in some respects also acts as a safety device, comparing its action here with that of a flywheel.

FLYWHEEL PUMPING ENGINES.

COMPARISON.

39. Although direct-acting steam pumps cannot be excelled in simplicity, low first cost, and small expense for repairs, yet they can never be extremely economical in their use of steam, even when built compound and triple expansion. While there is little doubt that a high-duty attachment will greatly increase the economy, the fact remains that at present only a limited number thus fitted are in use, and the above statement holds good for direct-acting steam pumps of the ordinary design.

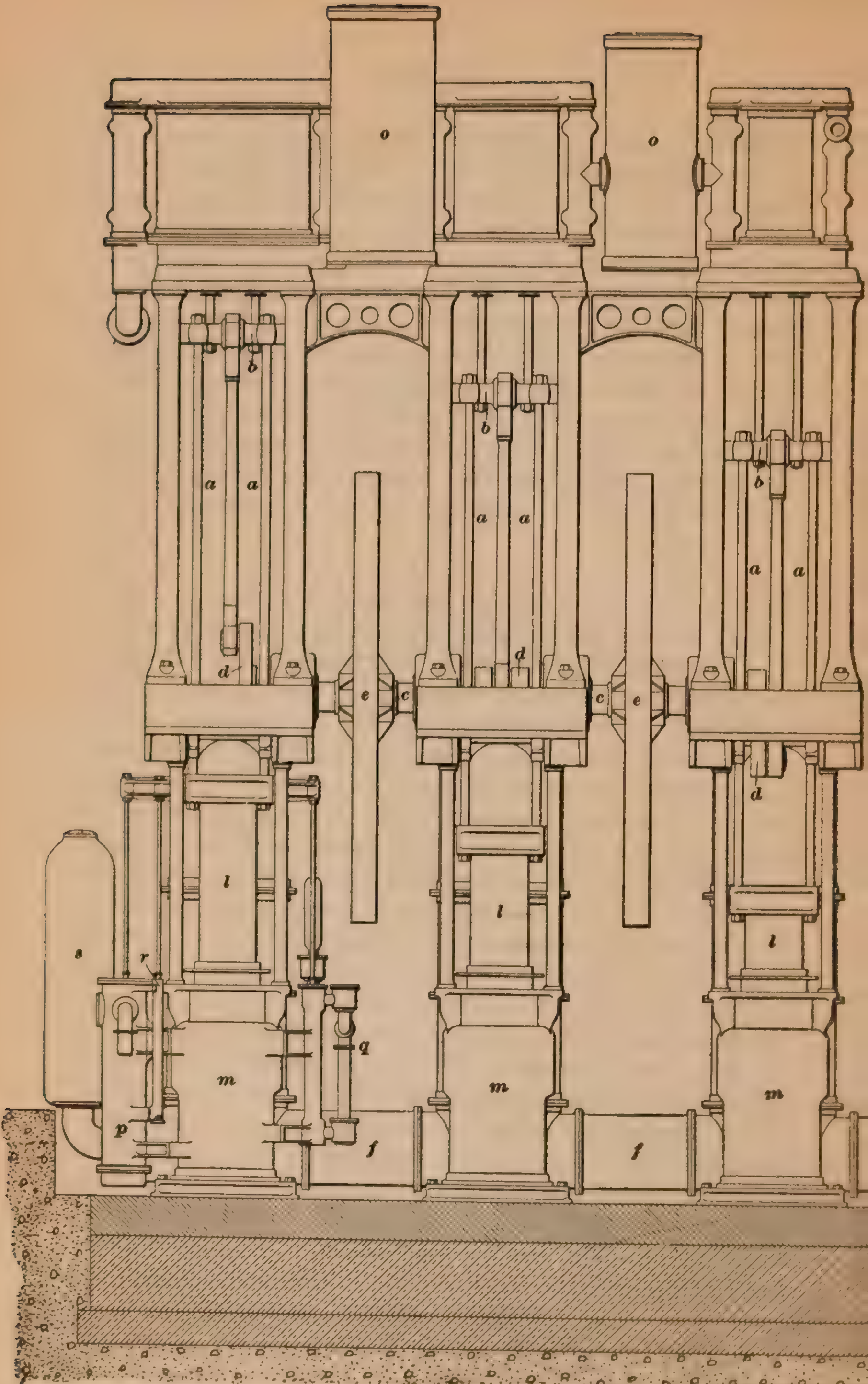
40. In large pumping stations and in many other cases where the cost of fuel is of more importance than the advantages gained from direct-acting pumps, flywheel pumping engines are often used. These are steam engines with cranks and flywheels usually designed for the particular purpose of driving the pump to which they are attached. The steam valves are driven in the ordinary way by means of eccentrics, or some approved automatic valve gear may be used to operate them. By the use of the flywheel, steam may be cut off at the most economical point in the stroke, and the surplus energy imparted to the steam piston during the first part of the stroke will be stored in the flywheel, to be given up towards the end, thus furnishing a nearly uniform driving force for the pump, piston, or plunger.

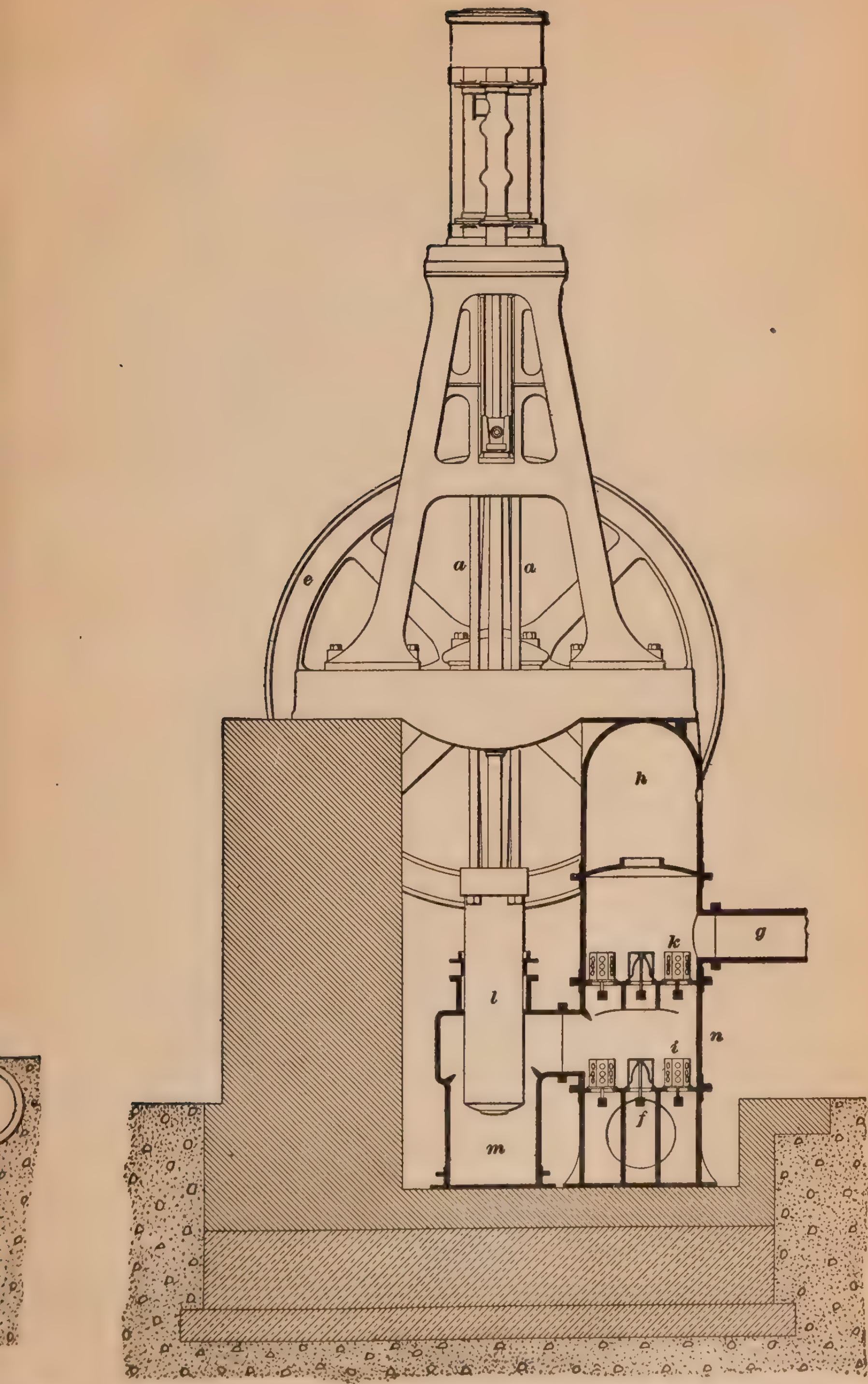
EXAMPLES OF FLYWHEEL PUMPING ENGINES.

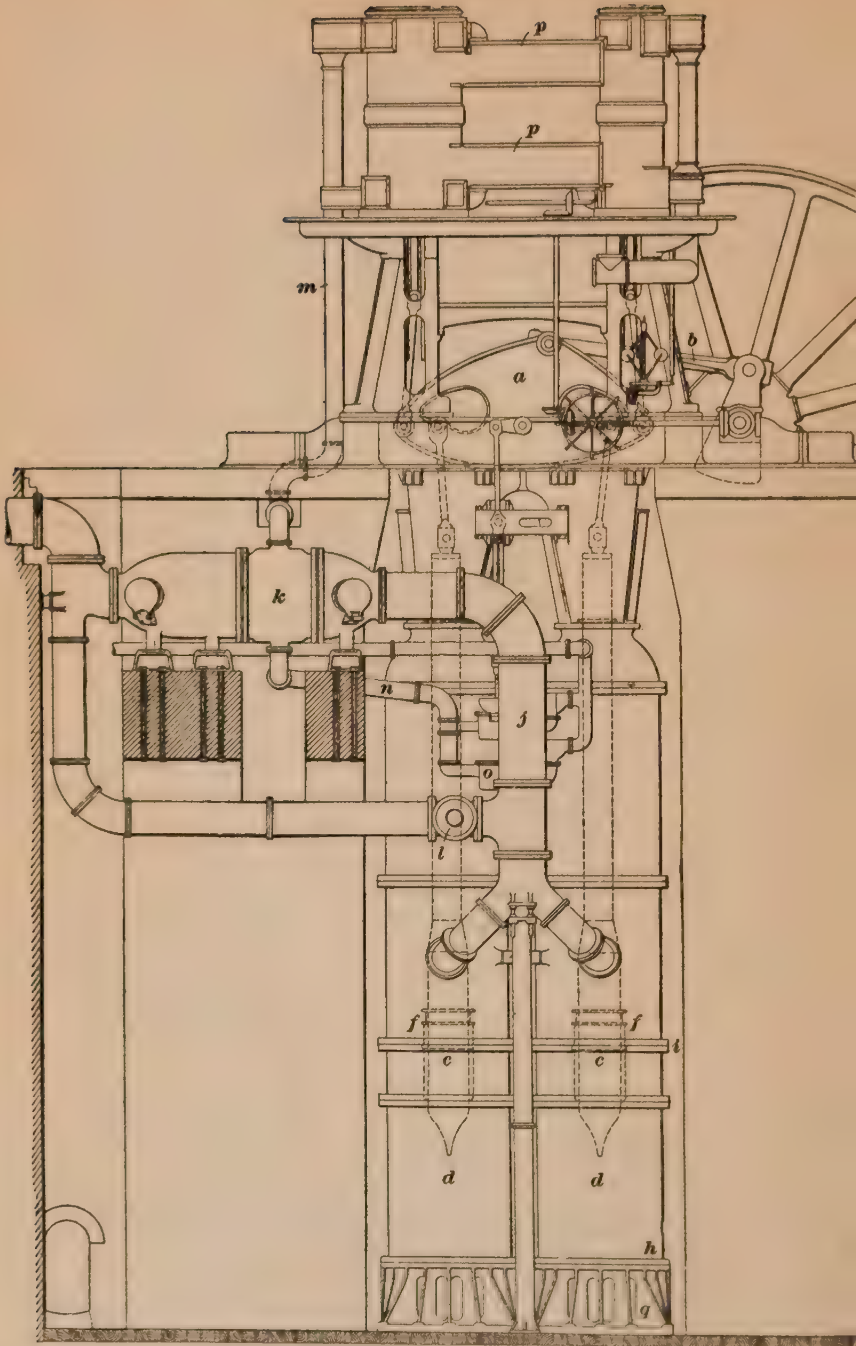
41. Fig. 13 shows a section of one side of a *Holley-Gaskill* compound pumping engine. The engine is double, the other side being like the one shown in the figure, the

two engines having a common flywheel and crank-shaft, with cranks set 90° apart. The high-pressure cylinder is placed directly over the low-pressure, with short passages between them. The connecting-rods from the two cylinders are attached to the opposite ends of a short walking beam *B*. By this arrangement the pistons move in opposite directions and the exhaust from the high-pressure cylinder passes directly to the low-pressure one. The valves are of the Corliss type, with a releasing gear for regulating the cut-off in the high-pressure cylinder. The connecting-rod that actuates the crank is attached to the upper end of the walking beam, and the rod that works the pump plunger *P* is attached to the crosshead of the low-pressure piston.

42. Fig. 14 is a front and side elevation of a modern high-duty triple-expansion pumping engine erected at the North Point pumping station, Milwaukee, Wisconsin. The engine is of the vertical inverted three-cylinder type, having the pumps in line with the cylinders, and is condensing, the condenser not being shown. Each piston is connected to a separate outside-packed single-acting plunger by means of pump rods, as *a*, *a*. There are four pump rods to each plunger, which are joined to the steam crossheads *b*, *b* and straddle the crank-shaft *c* in such a way as to allow the cranks *d*, *d* to rotate freely between them. Two flywheels *e*, *e* are employed to give uniform rotation to the machine. In the figure, *f* is the suction pipe; *g* is the delivery pipe, the delivery from each chamber being connected to a common delivery main not shown in the illustration; *h* is the air chamber; at *i* are the suction valves; at *k* are the delivery valves; *l*, *l* are the plungers and *m*, *m* the pump chambers; *n* is one of the valve chambers, the upper part of which forms the delivery air chamber *h* and also supports the fronts of the bedplates. The rear of the bedplates is supported on the masonry foundation. The steam cylinders are provided with Corliss inlet and exhaust valves on the high and intermediate cylinders and Corliss inlet valves

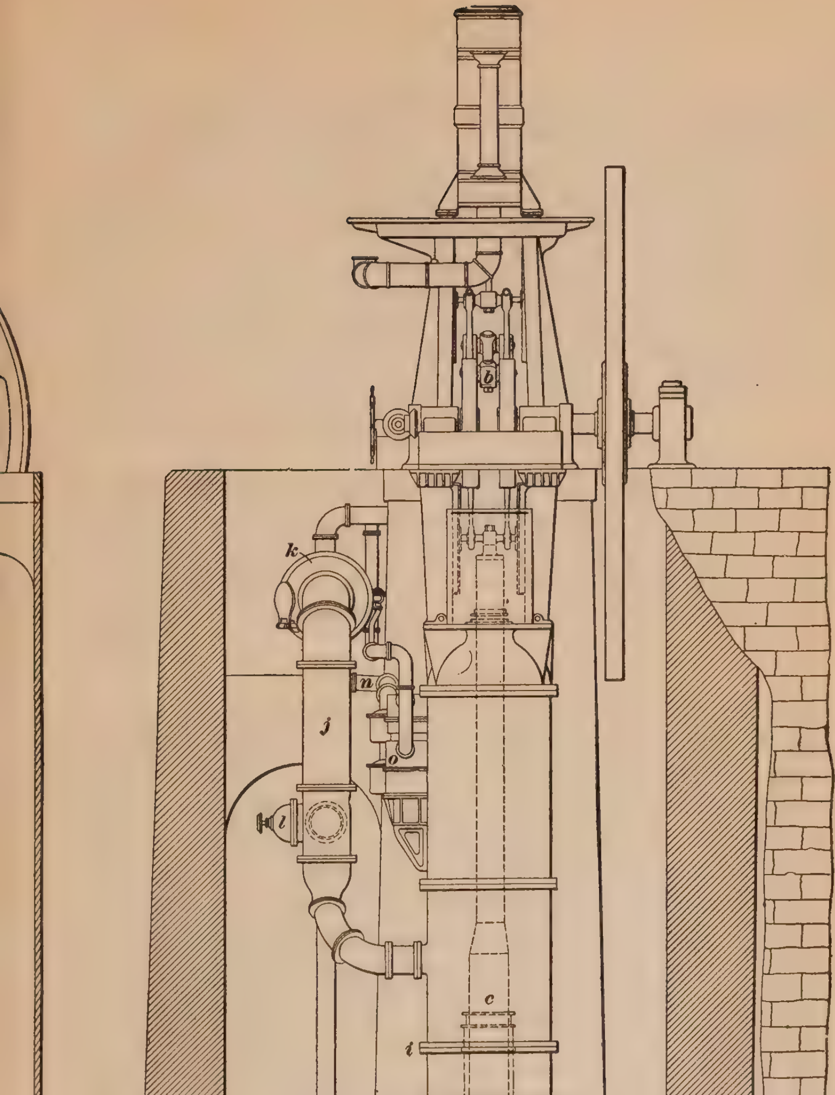

o
o
b
b
b
a
a
a
a
a
a
d
d
d
e
c
c
e
d
l
l
l
s
r
q
m
m
m
p
f
f


a
a
e
h
g
k
l
n
i
f
m



10B-291 § 34 T 6C § 38

(a)



(b)

and poppet exhaust valves on the low-pressure cylinders. Large reheating receivers *o*, *o* are used between the high and intermediate cylinders and between the intermediate and low-pressure cylinders. An air pump *p* is driven directly from the plunger crossheads and serves to remove the water of condensation, etc. from the condensers. An air-charging pump *q* pumps a small quantity of air into the water in order to replenish the air supply in the air chambers. A jacket drain pump *r* drains the water from the steam jackets. A suction air chamber *s* is fitted to the extreme end of the suction pipe and prevents shocks.

43. Pumps of the design shown in Fig. 14 are used almost exclusively for high-duty municipal water-works service and are extremely economical. This type of pump has given a duty as high as 160,000,000 foot-pounds of work done per 1,000,000 British thermal units supplied to the engine.*

44. Fig. 15 shows another type of high-duty municipal pumping engine, Fig. 15 (*a*) being a side elevation and Fig. 15 (*b*) the end elevation. This pump is of the crank-and-flywheel type; the motion of the pistons is not converted into a rotary motion in the manner shown in Fig. 14, but through the intervention of a rocking beam *a*, which is rocked back and forth by the high- and low-pressure piston and is connected to the crank and flywheel by the connecting-rod *b*. This design, from its designer, is known as the **Leavitt** design. Pumps of this type have rather more parts than the type shown in Fig. 14, but they are not so high and are more accessible. The pumps are of the plunger type and are inside-packed; in the illustration, *c*, *c* are the plungers, *d*, *d* the pump chambers, and *f*, *f* the inside plunger packings. The tops of the pump chambers form delivery air chambers. The suction valves are located

* The **duty** of a pump is a measure of its performance. It will be explained in detail later.

at *h* and the delivery valves are at *i*; the delivery pipe *j* discharges the water through the surface condenser *k*, thus using the delivery water for condensation. A butterfly valve *l* controls the amount of water passing through the condenser *k*. The exhaust pipe *m* from the low-pressure cylinder enters the top of the condenser; the pipe *n* leads from the condenser to the air pump *o*. This pump is double-acting and is driven from an arm attached to the beam *a*. Two reheating receivers *p*, *p* are used to heat the steam from the high-pressure cylinder during its passage to the low-pressure cylinder. The lower ends of the pump chambers rest directly on the bottom of the pump well, which is open to the river from which the pump takes its water. The water inlets are at *q* all around the base of the pump. It will be noticed by the arrangement of the connections of the steam piston and plungers to the beam that the steam pistons have considerably more stroke than the water plungers and consequently work at a considerably higher speed, which is a decided advantage in many respects. This pump, which is located at Louisville, Kentucky, gave the remarkable duty of 151,672,000 foot-pounds of work per 1,000 pounds of dry steam used by the engine, which is the highest duty on record for any compound engine.

ROTARY PUMPS.

45. Numerous attempts have been made to replace the reciprocating motion of the piston or plunger as used in the ordinary pump by a continuous rotary motion. The results have been unsatisfactory in many cases, owing principally to the difficulty in keeping the moving parts from wearing very rapidly, thus soon producing leakage.

46. Fig. 16 shows one of the oldest and at the same time one of the best **rotary pumps.** It consists of a chamber *a* in which two toothed wheels, or disks, *b*, *b* revolve in the direction shown by the arrows. The teeth of one wheel fit

accurately into the spaces between the teeth of its mate; and, as the wheels revolve, each tooth acts as a piston that pushes a certain amount of water ahead of it, thus drawing the water from the lower part of the chamber to the upper part, as shown by the arrows. It is very important that the flat faces of these wheels, or disks, should be a good fit between the cover and the bottom of the casing or cylinder, and the edges of the teeth also a good fit against the sides of the casing. Most of the rotary pumps that have been at all successful have been modifications of the form just shown, the principal difference being in the number and shape of the teeth on the rotating disks. One of these modifications is shown in Fig. 17. In this case the disk *a* has two teeth, or wings, which act as pistons, while its mate *b* has two recesses into which the teeth on *a* fit. The shafts of the two disks are provided with outside gearing that makes their relative motion positive and always keeps them in their proper relative position.

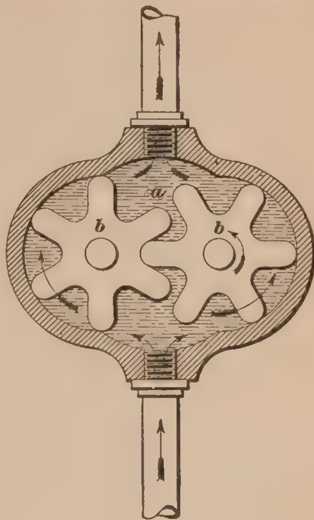


FIG. 16.

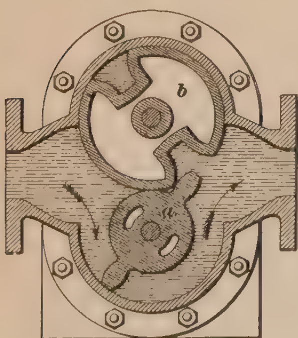


FIG. 17.

47. Fig. 18 is another modification of the rotary pump shown in Fig. 16 and gives a sectional view of **Root's cycloidal** rotary force pump. The shape of the disks or **impellers** *a*, *a* is such that the working surfaces when in contact roll upon each other. The sides of the casing are

semicircular and the impellers fit closely. The bearings in which the impeller shafts *b*, *b* run are adjustable in all

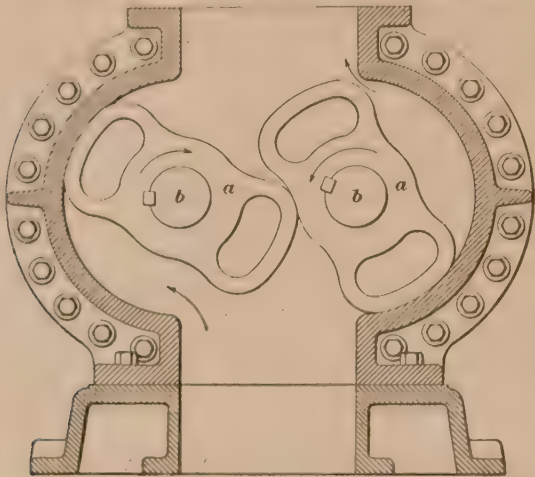


FIG. 18.

directions by means of wedges. This is claimed to be the simplest and most satisfactory rotary pump yet produced.

48. The **Quimby screw pump** shown in Fig. 19 is a rather peculiar form of a rotary pump. There are two

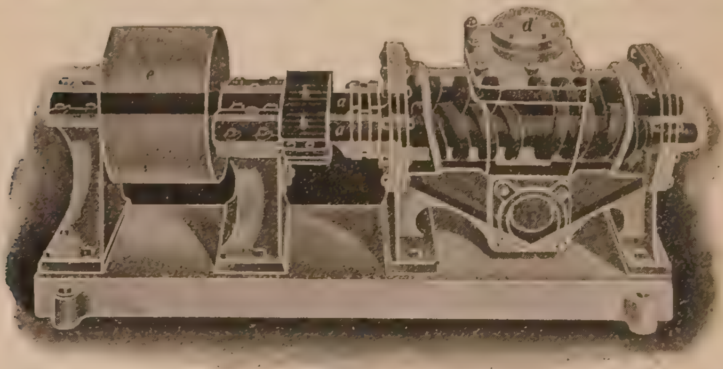


FIG. 19.

shafts *a*, *a* side by side and connected by the gears *b*, *b*. Each shaft carries a right-handed and a left-handed screw,

and the right-handed screw of one shaft meshes with the left-handed screw of the other shaft. The water coming through the suction pipe attached at *c* flows through passages in the casing to the outer ends of the screws and is drawn towards the center by the revolving screws, from whence it is discharged through *d*. The screws closely fit the pump casing and are a close running fit on each other. Since the screws are right-handed and left-handed and the course of the water is towards the center from the end of the four screws, there is no end thrust. The pump may be driven by a belt placed on the pulley *e*, or an engine or electric motor may be connected directly to it.

CENTRIFUGAL PUMPS.

49. Centrifugal pumps depend for their action on the pressure produced by the centrifugal force of a quantity of water rotated rapidly by the vanes of the pump. Fig. 20 shows two sectional views of a centrifugal pump and clearly shows its construction. The water flows through the suction inlet *a* into the chamber *b*, thus delivering the water to the inner ends of the vanes *c*, *c*, which revolve in the direction of the arrow. When the vanes are revolved, the air between them is driven out by centrifugal force, thus forming a partial vacuum. Water is forced in through the suction pipe by the pressure of the atmosphere and fills the space between the vanes. The water, of course, is made to revolve with the vanes, and the action of centrifugal force drives it outwards into the spiral-shaped passage *d*, which leads it to the discharge pipe connected to the outlet *e*.

50. Centrifugal pumps are most efficient when working under low heads and are seldom used for heads greater than 40 feet. For low heads and large quantities of water they give excellent results, and are especially useful when the water contains grit or other impurities that would destroy the pistons and packing or prevent the closing of the valves of other pumps. Since there are no valves or other

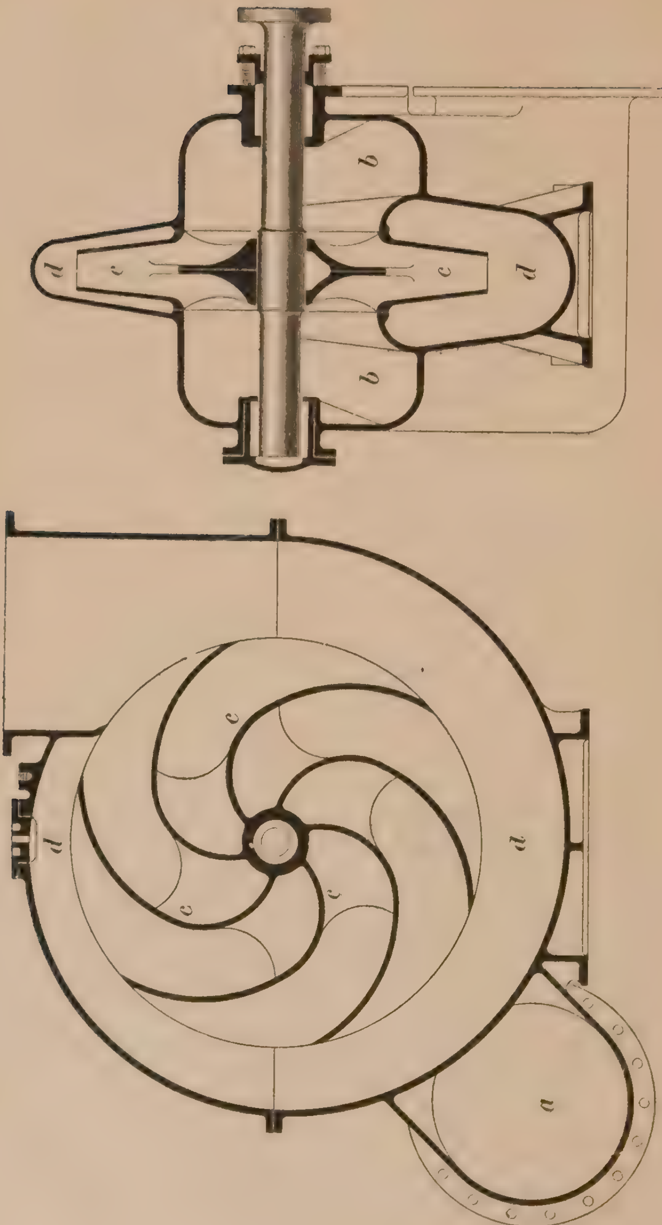


Fig. 20.

restricted passages, centrifugal pumps have been largely used in dredging machines for pumping water containing large quantities of mud, sand, and gravel; and, in fact, anything can be pumped that will be carried through the pump and pipes by a current of water. Centrifugal pumps may be belt-driven or be direct-connected to an engine or other motor.

POWER PUMPS.

51. Definition.—Pumps in which the piston or plunger is driven by a crank that receives its motion through a belt or gearing from some outside source of power are usually called **power pumps.**

52. Single Power Pumps.—A single power pump is one in which but one pump is driven by the shaft. This pump may be either *single-acting* or *double-acting.*

53. Duplex Power Pumps.—When two pumps are driven by cranks on a single shaft, the combination is called a **duplex power pump.** The discharge branches from the two pumps are generally combined in such a way that they discharge through a single pipe; and by a proper arrangement of the cranks, the flow through the discharge pipe and the power required to drive the pumps are made nearly constant. If the pumps are single-acting and the cranks are set 180° apart, the discharge from the two pumps will be the same as the discharge from one double-acting pump with the same diameter of piston and length of stroke. Duplex double-acting pumps, with cranks set 90° apart, are much used and give a very steady discharge, since, when one crank is on its dead center and its piston, consequently, is at the end of its stroke and momentarily at rest, the other piston is moving at its maximum velocity and discharging at its maximum rate.

54. Triplex Power Pumps.—Three pumps driven by cranks on a single shaft form a **triplex pump.** The most common application consists in the use of three single-acting plunger pumps with cranks set 120° apart. With

such a combination, at least one of the pumps is always discharging and one taking water from the suction pipe, and the flow is therefore continuous and nearly uniform.

55. Fig. 21 shows a type of triplex belt-driven power pump much used for feeding boilers, filling elevated tanks in

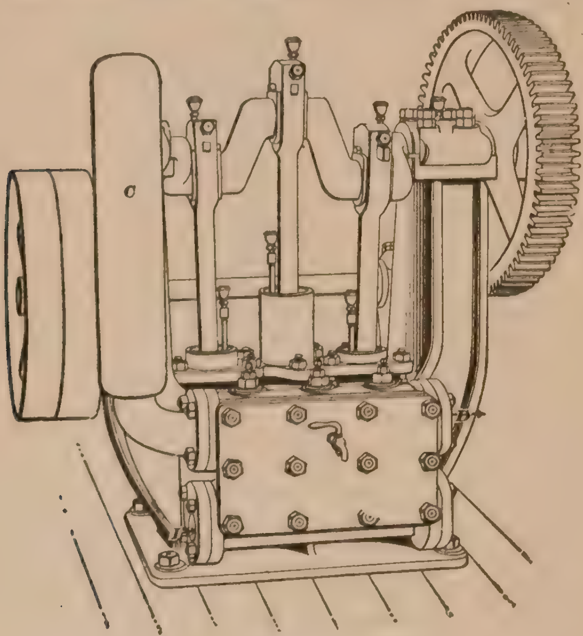


FIG. 21.

buildings, supplying hydraulic elevators, etc. It consists of three single-acting plunger pumps driven by cranks set at 120° on a single shaft. A tight and a loose pulley provide the means for starting and stopping the pump, without disturbing the engine or main shaft. The pulley shaft is geared to the crank-shaft by a pinion and spur wheel. *I* is the suction inlet, *D* the discharge opening, and *C* the air chamber.

56. Where the supply of power is steady, a belt-driven power pump is very convenient and economical for the purposes for which such pumps can be used, since they get their

power with the same degree of economy as the engine by which they are driven; they are also simple in construction and easily operated.

57. In locations where there is no steam or other power directly available, or where the use of the pump is so intermittent that a steam plant will not be economical, or where the cost of supplying steam is too great, power pumps driven by electric motors may be used to advantage. Small pumps driven by windmills, hot-air engines, gas engines, etc. are much used for supplying water to buildings that have no connection with public water works. Small, single-acting plunger pumps are most commonly used with these methods of driving, although double-acting pumps are sometimes used. Where water-power is available, pumps for city water works or for supplying manufacturing establishments are often driven by waterwheels.

MINE PUMPS.

SERVICE.

58. Pumps intended for the drainage of mines are probably subjected to the hardest usage of any. The water to be pumped is generally gritty and frequently it contains a large percentage of acids; a very high pressure must generally be pumped against and the pump has to run almost continuously for long periods at the full limit of its capacity. In most cases the mine is located quite remote from supplies; the pump of necessity is underground and in a rather limited space; it is generally of vital importance that the pump be kept running in order to prevent the drowning out of the mine, and for the same reason it is desirable that all wearing parts be very accessible so that repairs can be made in the shortest time. Furthermore, it is desirable that the pump continue at work even when covered entirely with

water. The exigencies of the service have led to designs of pumps especially suited for the work. While they do not differ essentially from ordinary pumps, they have generally a different arrangement of water end. Nearly all mine pumps are of the plunger pattern, the plunger pump, by reason of the ease with which leakage can be stopped, being best adapted for high pressures.

59. Mine pumps are either pit pumps, direct-acting steam pumps, or power pumps. By a pit pump is meant a pump having its water end located at the bottom of the mine and connected to a steam engine or other motor at the surface by rods. Pit pumps are the oldest type of mine pump and are still used to some extent.

TYPES OF MINE PUMPS.

CORNISH PUMPING ENGINE.

60. Until within comparatively recent times, the so-called **Cornish pumping engines** have been the only ones used for removing the water from the mines. This engine was invented by Watt for use in the mines of Cornwall and was the first really effective steam engine made. An illustration of a Cornish pumping engine is shown in Fig. 22. The cylinder A is single-acting; that is, the steam acts only on one side of the piston. The piston rod B is connected to the walking beam C by a link R. In Cornish pumping engines, the steam is admitted through the valve in I to the top of the piston and forces it down towards the bottom of the cylinder. The weight of the pump rods and other moving parts in the shaft, which parts are called the **pit work,** is sufficient to raise the piston to the top of the cylinder when the steam on the upper side of the piston is put in communication with the lower side. The cylinder A is steam-jacketed; that is, the cylinder walls are hollow and

filled with steam in a manner similar to the water-jacket of an air compressor, the steam entering through the pipe *K*.

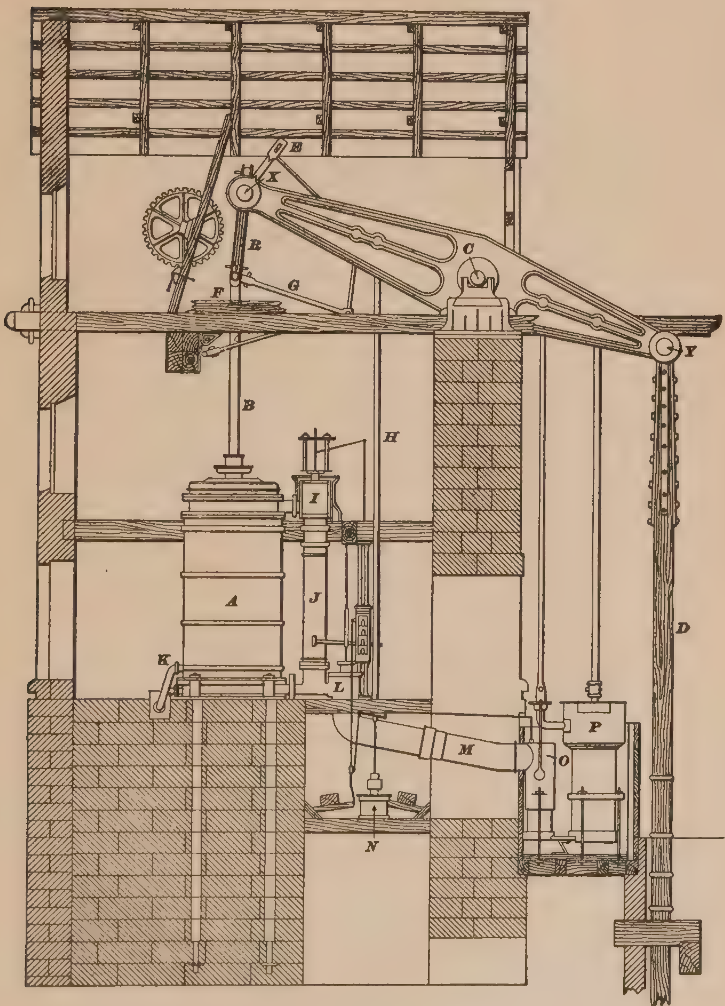


FIG. 22.

61. The action of the pump is as follows: Steam is admitted to the upper side of the piston through a valve in *I*,

which is operated by means of a tappet rod *H*. The steam is of high pressure and forces the piston rod downwards and at the same time raises the pit work. This gathers momemtum while coming upwards, and the steam is cut off, expanding during the rest of the stroke. Just before the end of the stroke, what is termed an *equilibrium valve*, also located in the casing at *I*, opens and allows the steam in the upper end of the cylinder to communicate with that in the lower end. The two pressures being thus balanced, the heavy pit work causes the right end of the walking beam *C* to descend, raising the piston to the top of the cylinder again. The exhaust valve is located at *L*. When this is raised, the exhaust steam flows through the pipe *M* into the condenser *O*. *P* is a small pump used in operating the condenser. *E* is a catch intended to act in case the valves should fail to work. The piston rod passes between two blocks, of which *F* is one, the other being opposite. If the left end of the walking beam should descend too far, a crosspiece on the catch rod *E* is caught by the blocks *F* and prevents any further downward movement of the piston.

BULL ENGINE.

62. In Fig. 23 are shown two views of a Cornish Bull engine and pump. This style of pumping engine is made by many firms and differs but very little in regard to details. Here the walking beam is dispensed with and the cylinder is placed directly over the shaft, the pit work being attached to the piston itself. In this case also the cylinder is single-acting, the steam being admitted below the piston instead of above it, as in the engine described in Fig. 22. The condenser is usually omitted in this class of pumps, the steam exhausting directly into the atmosphere. In case the weight of the pit work should be greater than necessary to force the water up the required height, the extra weight is counterbalanced.

63. The Bull pumping engine possesses several advantages over the Cornish pump. The heavy walking beam

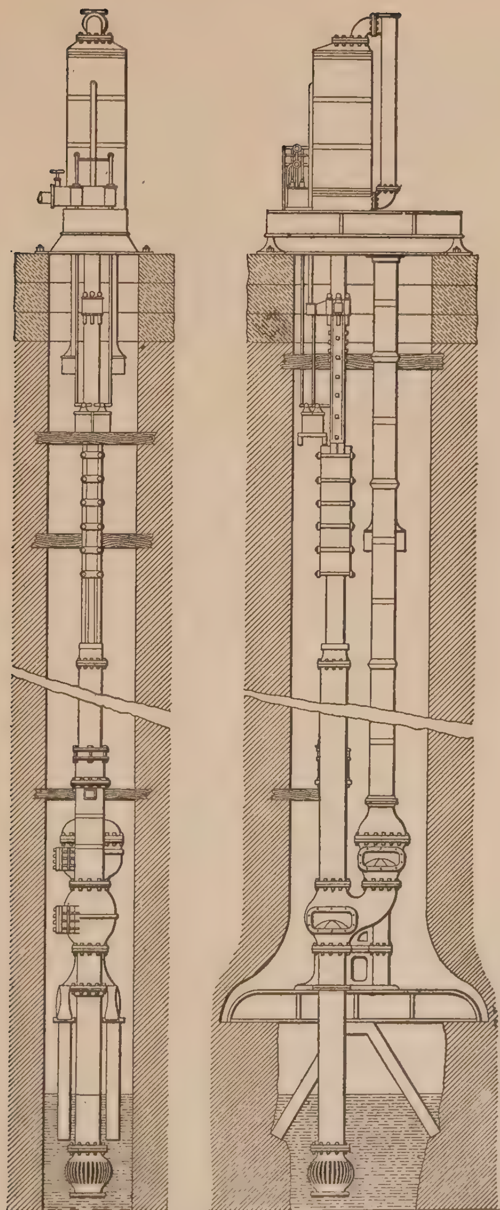


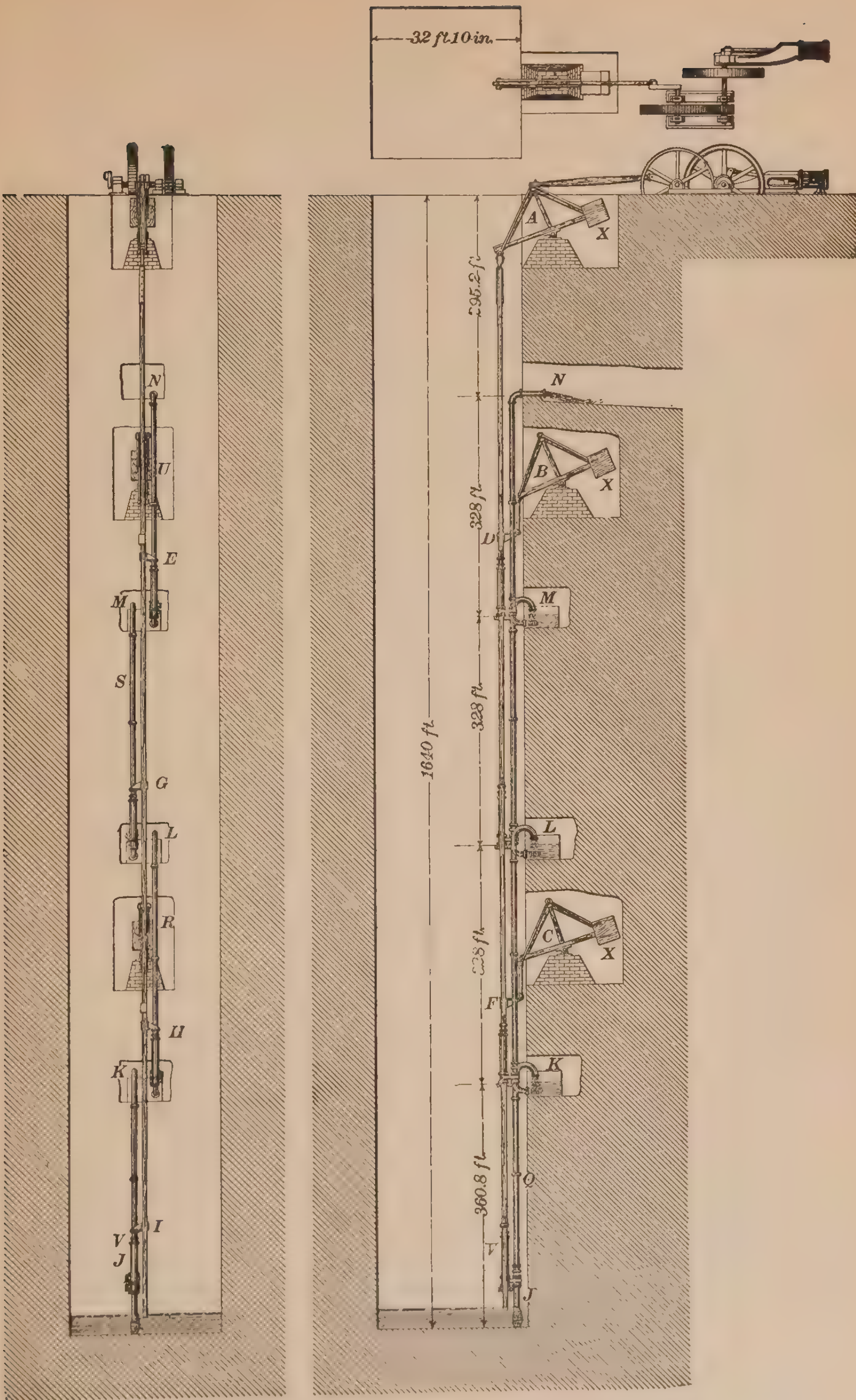
FIG. 23.

and its connections are dispensed with; this lessens the first cost; the friction is greatly reduced; the advantage of having a direct-acting engine is also obtained. The principal disadvantage is that the pump being directly over the shaft, takes up a great deal more room where space is necessary than the Cornish pump.

64. Cornish and Bull pumps both use steam expansively. They do not have flywheels to absorb the energy of the early part of the stroke and give it out again at the end, but utilize the heavy pit work to accomplish the same purpose. The number of expansions ranges from four to ten; that is, the steam is cut off from $\frac{1}{4}$ to $\frac{1}{10}$ stroke. When using more than six expansions ($\frac{1}{6}$ cut-off), the strain produced on the machinery becomes very heavy, and the resulting wear and tear of the machinery more than makes up for the increased economy in the use of steam. Many engineers claim that four expansions are the most economical.

PIT-PUMP ARRANGEMENT.

65. The arrangement of the pump driving mechanism, pit work, and balancing mechanism in a deep mine shaft is shown in Fig. 24. In this case an ordinary horizontal engine is used at the surface, which works the pumps through the intervention of a gear train and a connecting-rod and crank, the connecting-rod being attached to the bell-crank *A*. On account of the great length of the rod (over 1,600 feet), its weight added to the weight of the plungers is considerably more than the weight of the water column; hence, to save the extra power which would be required to be used in raising this extra weight, it is counterbalanced. A counterbalance weight *X* is placed on one end of the bell-crank *A*; two other bell-cranks, *B* and *C*, are located down the shaft, one end carrying the counterweight *X* and the other end being connected to the pump rod by means of a link and the cast-iron offsets *D* and *F*. The water is raised by four lifts, the first, to *K*, being 360.8 feet, and the other three 328 feet each. In this particular instance, the water is



32 ft 10 in.
N
U
E
M
S
G
L
R
H
K
I
V
J
A
X
295.2 ft
N
B
X
328 ft
D
M
328 ft
1640 ft
L
C
X
328 ft
F
K
360.8 ft
O
V
J

Fig. 24.

discharged into a tunnel *N*, about 300 feet below the surface. The pump rod goes straight down the shaft and the discharge pipes are placed alternately on each side of it. *J* is a suction pipe. *I* is a bracket, one end of which is attached to the pump rod and the other end to the pump plunger *V*. On the down stroke, the water is forced out of the pump cylinders and up the pipes *Q*, *R*, *S*, and *U*, discharging at *K*, *L*, *M*, and *N*. The same pit work and pump arrangement may be and is used for Cornish and Bull pumps.

66. The use of a geared engine possesses several advantages over the Cornish or Bull pumping engines. The flywheel permits a more even distribution of the power. The length of the stroke is always the same, and there is no danger of damage caused by the piston being blown through the cylinder head, should the valve gear refuse to work.

WATER END OF PIT PUMPS.

67. Comparison of Lifting and Force Pumps.—The water end of a pit pump may be a lifting pump or a force pump. The lifting pump is generally considered inferior to the force pump (which latter is almost invariably of the plunger pattern) for mine work.

It is easier to specify the objections to lift pumps than to state their advantages over the plunger pumps. The pump rod, being necessarily inside of the delivery pipe, reduces the effective area of pipe and increases the friction of the water to some extent, owing to the added surface rubbed against. The rods are concealed and cannot be inspected without removing the entire rod. Not only do the bolts and rods sometimes break, thus rendering their recovery difficult, but the bolts will wear against the stocks, causing loss of power by friction and destroying the pipes. Lift pumps are not so liable to sudden injurious strains as the plunger pumps.

The plunger type of pumps is superior to the lift pump in nearly every respect for very high lifts with the accompanying heavy pressure or when dirty water is being raised.

When pumping against a heavy pressure, it is impossible to keep the piston of lift pumps tight and prevent the water from leaking. The piston and cylinder of the lift pump must in every case be a perfect fit and be truly cylindrical. With plunger pumps, on the contrary, the rod passes through a stuffingbox, and the plunger may or may not fit the cylinder. When pumping dirty water, the grit comes in contact with the surface that the piston of a lift pump is constantly traveling over and destroys both the cylinder and piston very rapidly; whereas, the plunger has to be kept tight at only one permanent place, and the dirt cannot very well get at the surface of the packing on which the plunger or plunger rod rubs. Every part of a plunger pump can be readily examined and repaired without being obliged to take down the whole apparatus.

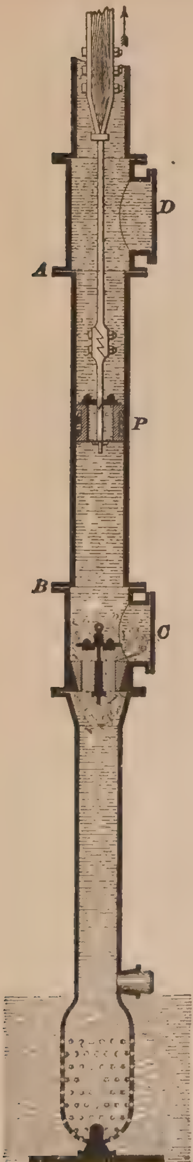


FIG. 25.
291—31

68. Example of a Lifting Pump. In Fig. 25 is shown a section of a lifting pump for use in mines. The pump consists of a series of pipes connected together, of which the lower end only is shown in the figure. That part of the pipe included between the letters *A* and *B* forms the pump cylinder in which the piston *P* works. The part above the highest point of the piston travel is the delivery pipe, and the part below the lowest point of the piston travel is the suction pipe. When speaking of these parts as applied to mine pumps, the delivery pipe is usually termed the **working barrel,** and the suction pipe the **wind bore.**

In mine pumps, the lower end of the wind bore is pear-shaped and perforated

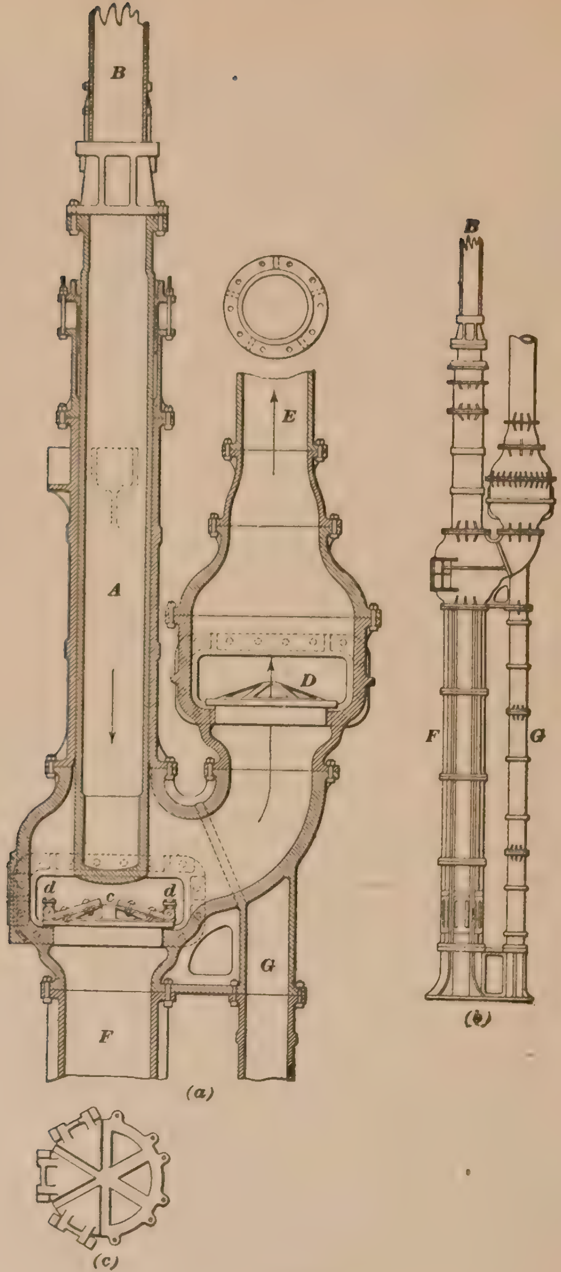


FIG. 26.

with many small holes to keep solid matter in the water from entering the pump and destroying the valves. In some cases, the pear-shaped end is covered with gauze for the same purpose. A bonnet *C* may be removed to allow the suction valve to be repaired, and a bonnet *D* gives access to the piston and its valves. The pump rod is made of wood strapped with iron and is connected to the piston in the manner shown by the illustration.

69. Example of a Force Pump.—Fig. 26 shows one design of a force pump of the plunger type as used for a pit pump, Fig 26 (*a*) being a section showing the pump cylinder and valves, and Fig. 26 (*b*) showing an elevation of the whole water end drawn to a smaller scale. The plunger *A* is hollow, the weight of the heavy rod *B* and connections being sufficient to raise the water to the required height.

Suppose the plunger to be on the down stroke; the valve *c* is then closed and the water filling the pump cylinder is forced through the valve *D*, which it opens, and up the delivery pipe *E*. When the plunger reaches the end of its stroke and begins its return, the weight of the water forces the valve *D* to its seat, retaining the water above it in the discharge pipe *E*. As the plunger moves upwards it leaves a partial vacuum behind it, causing the water to rush up the suction pipe *F*, lift the valve *c*, and fill the pump cylinder. The plunger makes another downward stroke and the above process is repeated. A support *G* is attached to the delivery pipe, the lower end resting on a foundation. This is necessary, since the great weight of the water in the discharge pipe and the weight of the pipe itself would break it off at the bend unless supported in some such manner; otherwise, the thickness of the metal around the bend would necessarily be enormous.

70. A top view of the valves is shown in Fig. 26 (*c*). They consist of six triangular valves arranged in a circle, with their apexes pointing towards the center. These six valves turn upwards on hinges and are prevented from going too far by the projection *d*; see Fig. 26 (*a*). Three

of the valves have been removed so as to show the amount of valve opening that they give. When the valves are open, they form an angle of about 45° with their position when closed.

SINKING PUMPS.

71. Purpose.—When putting down a new shaft or deepening an old one, the so-called **sinking pump** is used to drain the water from the shaft bottom so that the work may proceed. These pumps must necessarily be portable and are suspended by a chain attached to eyebolts in the pump. They are also provided with wrought-iron clamps, by means of which they may be attached to the timbers in the shaft when it is desired to fix them in position temporarily. Hence, as the shaft gets deeper, the chain may be lengthened out, an extra joint placed on the upper end of the delivery pipe, and it is again ready for business. The sinking pump is subjected to the hardest usage of any mine pump. The water pumped is invariably gritty and often acid. The water trickling down on the pump from above carries mud along with it and so completely covers the pump that it is hardly distinguishable at times from the soil itself. Notwithstanding all this, a sinking pump must work night and day, often up to the limit of its capacity, and its failure, even for a day, at a critical period may flood a shaft which would require a week or more to recover.

72. Steam Sinking Pump.—In Fig. 27 a Cameron sinking pump is illustrated. This pump meets all the conditions required of a sinking pump and is a favorite with mine operators. There is no outside valve mechanism whatever, and nothing short of actual breakage of the pump itself or of the steam, suction, or delivery pipe can prevent the pump from working. The manner of suspending it from a chain is shown in the illustration, also the method of attaching it to the shaft timbers. In order to more clearly show the working of the valves and plunger, a partial section

of the pump is given in the figure. The pump has one plunger, but is double-acting by reason of its peculiar construction. It will be noticed that leakage past the plunger *A*

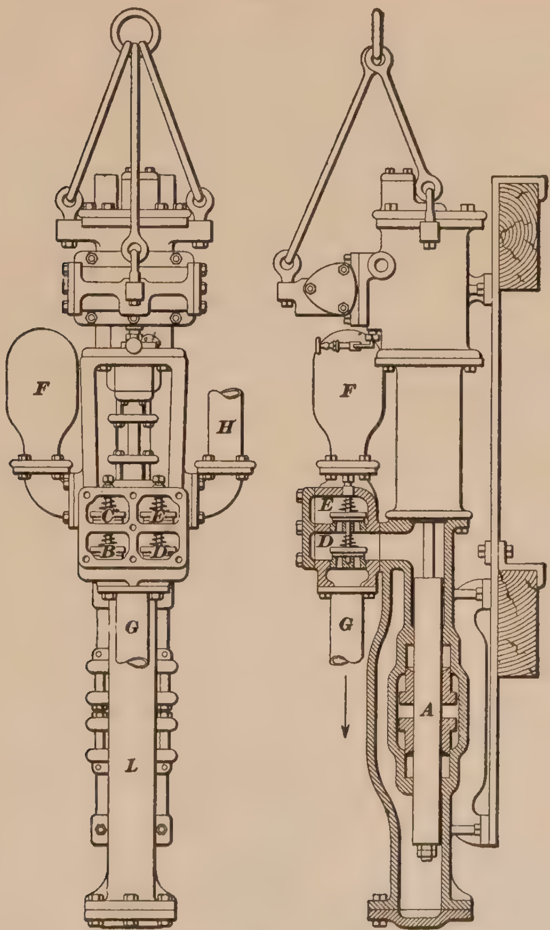


FIG. 27.

is prevented by two stuffingboxes and glands placed in the center of the pump cylinder; a pump having this arrangement is said to be **center-packed.**

The action of this pump is as follows: Suppose the plunger to be moving downwards. The water is forced out of the chamber *L*, which communicates with the delivery pipe *H* by means of the valve *C*, and lifts *C*, thus flowing into *H*. As the plunger moves down it leaves a vacuum behind it; the water in the shaft rushes up the suction pipe *G*, raises the valve *D*, and fills the upper part of the plunger cylinder. When the stroke is reversed, the valves *C* and *D* close, and the valves *E* and *B* open, the water being forced up the pipe *H* through the valve *E*, and the chamber *L* is filled through the opening of the valve *B*. *F* is the air chamber. The section shown by the view on the right is taken in a rather peculiar manner, the greater part being taken through the center line of the engine so as to show the plunger, stuffingboxes, etc., and the part showing the valves being taken on the center line of the valves *E* and *D* of the view on the left.

73. Electric Sinking Pump.—While most sinking pumps are steam-operated, electrically driven sinking pumps are also used. Fig. 28 shows a duplex electric sinking pump of the center-packed type, the stuffingboxes being shown at *H*. The two plunger rods *E* and *F* operate the plungers. A clamping piece *D* is used for attaching the pump to the shaft timbers; an eyebolt *G* is used for suspending the pump from a chain. The water enters through the suction pipe *B* and leaves through the discharge pipe *A*.

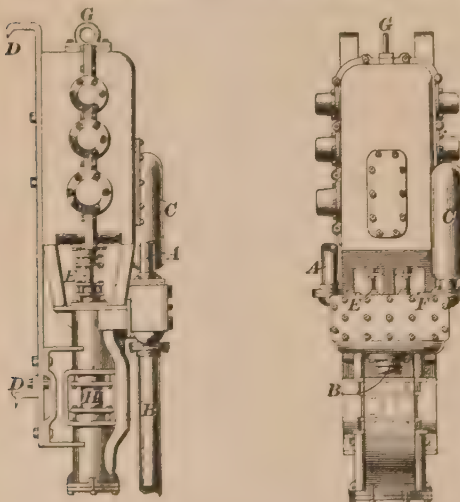


FIG. 28.

An air chamber C is fitted to the valve chamber. The electric motor is within the water-tight casing above the water end and is protected by it, so that the pump can work just as well under water as above it.

DIRECT-ACTING STEAM PUMPS FOR MINE WORK.

74. Pumps Used.—Direct-acting steam pumps used for mine drainage are almost invariably of the plunger pattern. Most of them are duplex, but a number of single double-acting steam pumps are in use. Formerly, all the mine steam pumps were simple direct-acting pumps, but of late years compound and even triple-expansion pumps have grown in favor, and even crank-and-flywheel pumps driven by compound Corliss engines are now extensively used on account of their superior economy. Most of the pumps are of the *double-plunger type*, there being two plungers to each water cylinder, and the stuffingboxes are located on the outside, thus making the pumps *outside-packed*. Some mine pumps are *center-packed* and use but one plunger for each water cylinder.

75. Simple Double-Plunger Pump.—Fig. 29 shows a side view of a simple direct-acting single mine pump of

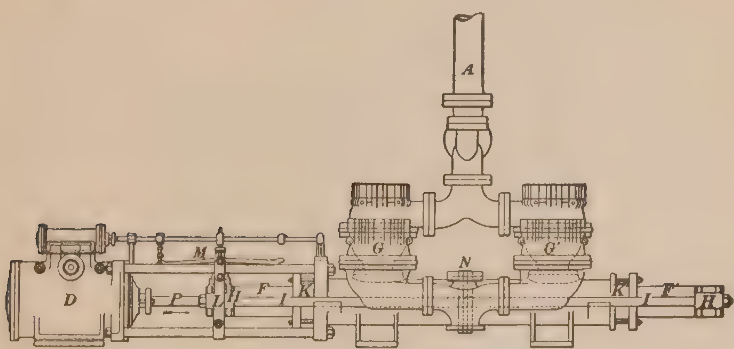


FIG. 29.

the double-plunger type. The two plungers F and F' carry yokes H, H at their outer ends and are tied together by

side rods, as I. The plunger F is attached directly to the piston rod P. Suppose the steam piston in D to be moving to the right; the plunger F is then forcing water into the chamber G and up the discharge pipe A. Since the plunger F' is moving out of the water cylinder (it will be understood that the cylinders in which F and F' work are divided by a water-tight partition at N), water flows in through the suction pipe, and when the pump makes its return stroke, F' does the forcing while water flows into the cylinder in which F works. It is thus seen that by the use of two plungers connected as shown, the pump is made double-acting. Stuffingboxes K and K' are used for packing the plungers.

76. Compound Double-Plunger Pump.—The internal arrangement of a double-plunger pump is shown clearly in Fig. 30, which is a sectional view of one side of a Jeanesville compound duplex mine pump designed for heavy pressures. The section shows one of the plungers E working inside its chamber; G is the partition that separates the two chambers. The outer end of each plunger is supported by a shoe K working on a slide J. L is the suction pipe with branches leading to the suction valve chambers F, F; the discharge valve chambers H, H connect with the discharge pipe shown just over the pump cylinders. As shown in the figure, there are two suction and two discharge valves for each plunger. The usual arrangement for pumps of this size is to have a great number of small valves instead of a few large valves, as shown, but for mine work the sulphur in the water destroys the valves rapidly and the large valves are more quickly and cheaply replaced. A is the main and B the auxiliary throttle; C is the high-pressure cylinder, from which steam goes to the low-pressure cylinder through the pipe D. The valve gear of the steam end is practically the same as that of Worthington pumps, and the steam valves of one side are operated from the piston rod of the other side. Steam is carried full stroke in all cylinders.

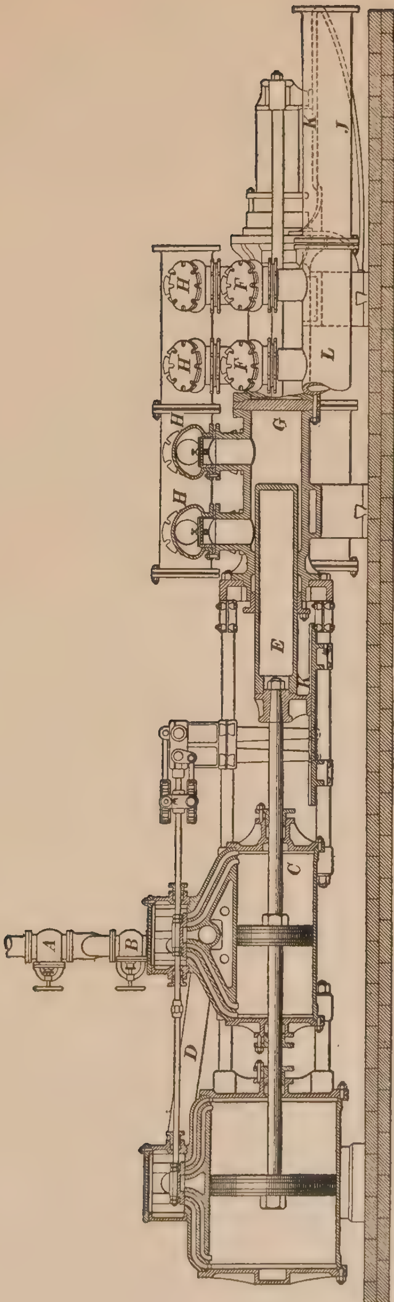


FIG. 30.

77. Triple Expansion Center-Packed Pump.—Fig. 31 is a side view of one side of a triple-expansion duplex Worthington mine pump having plungers which are center-packed.

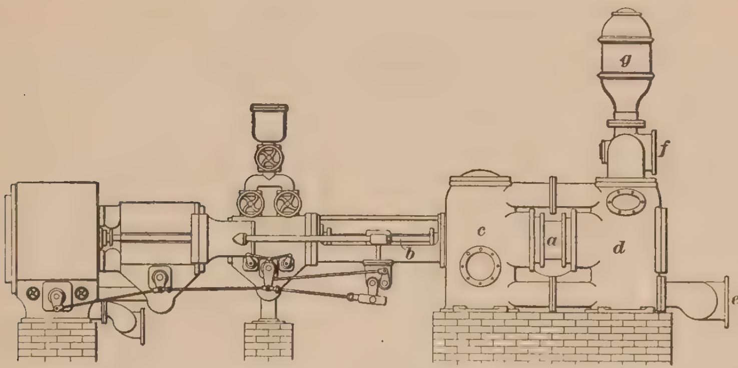


FIG. 31.

The type of water end used with this pump has been given the name of **Scranton type** by the makers. Sectional views of the steam cylinders of this pump have already been given in Figs. 9 and 10. The plunger *a* is connected to the piston rod *b* and works in the pump chambers *c* and *d*, which have the suction valves on the bottom and the delivery valves on top. The suction pipe is connected at *e* and the delivery valve at *f*. An air chamber *g* on the delivery absorbs shocks and promotes a steady delivery. The pump is double-acting.

FLYWHEEL PUMP.

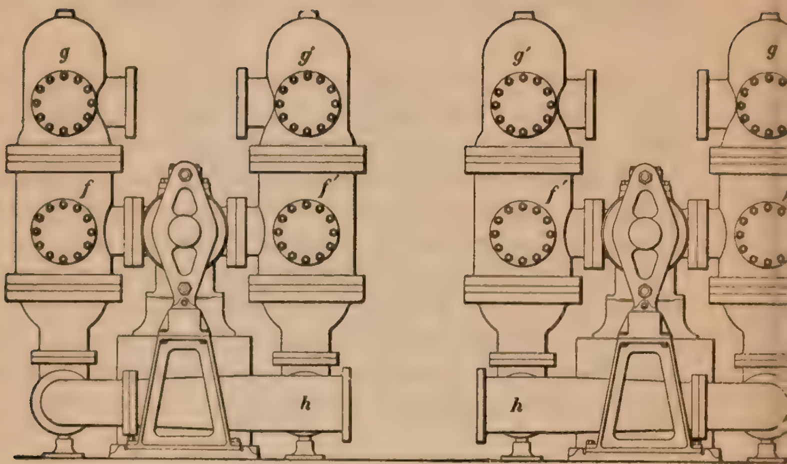
78. Fig. 32 (*a*) is a side view of the high-pressure side of a duplex pump driven by a cross-compound Corliss engine, the pump being of the double-plunger type. Fig. 32 (*b*) is an end view of the water end of both pumps, looking towards the engine, and Fig. 32 (*c*) is an end view of the engine, looking towards the flywheel, the observer being supposed to stand between the pumps and the engine. The plungers *a* and *b* are connected by yokes *c*, *c* and rods *d*, *d* and are driven

directly by the piston rods of the high-pressure and low-pressure cylinders, which for this purpose are prolonged beyond the pistons and pass through the back cylinder heads.

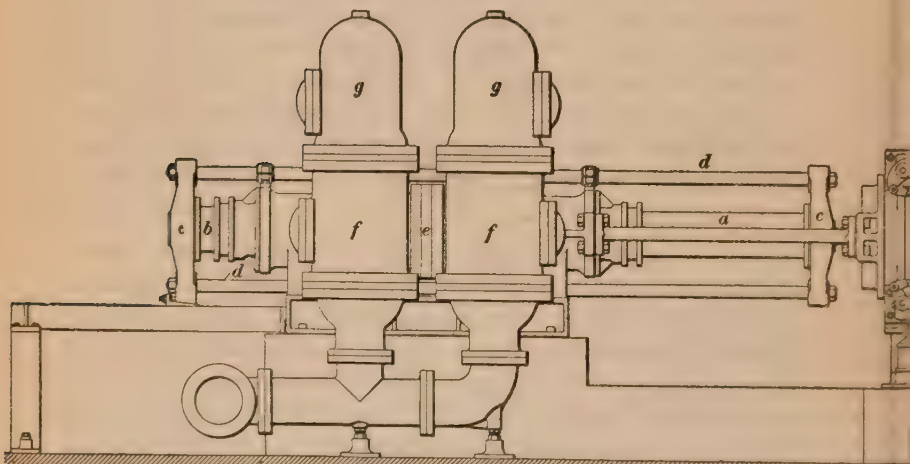
79. The pump cylinders have the necessary diaphragm *e* in the center, and each pump cylinder has two valve chambers *f*, *f′* containing the suction valves and two valve chambers *g*, *g′* containing the delivery valves. These valve chambers are placed on both sides of the pump cylinders. The four suction-valve chambers of each pump connect to the common suction branch *h*, and the two branches in turn are connected to the suction main by a **Y** fitting not shown in the illustration. The four delivery chambers of each pump are connected together by branch pipes, and these branch pipes in turn discharge into a common main delivery pipe.

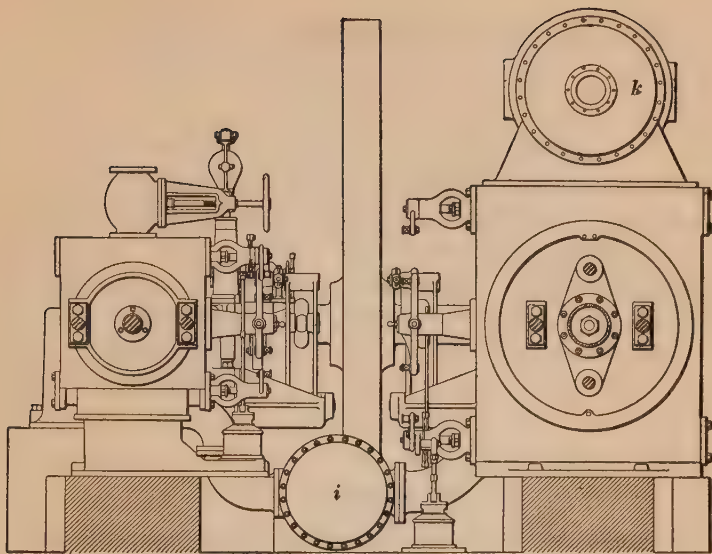
80. A reheating receiver *i* is placed between the high-pressure and low-pressure cylinders. The low-pressure steam-inlet valves are placed beneath the low-pressure cylinder; the low-pressure exhaust valves are on top and exhaust directly into the condenser *k*, which is placed on top of the low-pressure cylinder. The high-pressure valves are arranged in the usual way. The engine is provided with a variable speed Porter governor *l*, by means of which the speed of the engine may be varied to suit the requirements of the service.

81. The particular pump illustrated has cylinders 32 inches and 60 inches in diameter and a 48-inch stroke, the plungers being 13¾ inches diameter. It was designed by the Dickson Manufacturing Company, of Scranton, Pennsylvania, to pump water highly charged with sulphuric acid against a head of 700 feet. To guard against corrosion, the pump cylinders, valve chambers, and all pipes were lined with lead and the plungers and valves made of acid-resisting composition. Owing to the high economy possible through the use of a compound condensing Corliss engine, this is fitly called a "high-duty mine pump" by the builders.

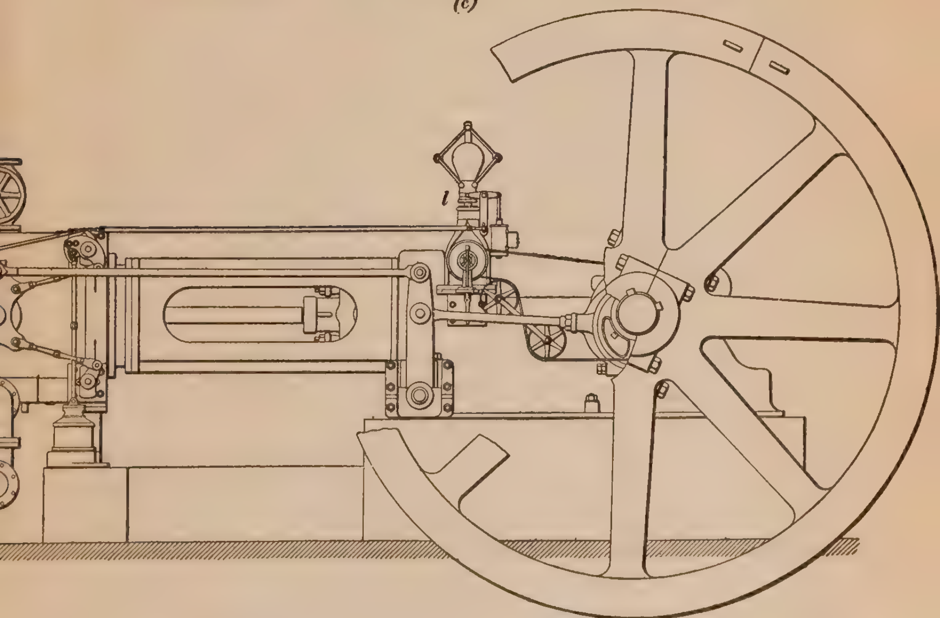


(b)





(c)



2

DISPLACEMENT PUMPS.

DEFINITION AND CLASSIFICATION.

82. A **displacement pump** is a pump in which there is a complete absence of moving parts and where the fluid to be pumped is moved by steam or compressed air. Of the steam-operated displacement pumps, the best known is the *pulsometer;* the *Harris compressed-air direct-air-pressure pump* and the *Pohlé air lift* are the best known air-operated displacement pumps.

THE PULSOMETER.

83. Fig. 33 shows a perspective view and Fig. 34 a sectional view of a pulsometer of the latest manufacture. In the sectional view the full lines represent the left-hand half and the dotted lines indicate the position of the discharge valves in the right-hand half of the pulsometer shown in Fig. 33. In the following description, the letters refer to both figures: The steam pipe is connected at *E* and the suction pipe at *S*. *C* is an air chamber that has no connection with *B* and *A*, but communicates with the suction pipe by means of the opening *I* situated below the suction valves *F* and *G*. The two latter valves are made of flat rubber and are held to their seats, as shown in the figure, by means of the spindles *R* and *T*. The spindles are raised and lowered, as the case may require, by means of the bolts *f* and *e*. *H*, *H* are plates that may be removed to facilitate the examination of the valves. *D* is a hard-rubber ball that acts as a valve for admitting the steam to the chambers *A* and *B*. *M* and *N* are exhaust valves, also made of rubber and situated in the chamber *L* attached to the other half of the cylinder. They are raised and lowered in the same manner as the suction valves by turning the bolts *g* and *h*. *K* is the delivery or column pipe.

84. The action of the pulsometer is as follows: Both chambers *A* and *B* are filled with water to about the height of the water in *B*, Fig. 34. The valve *d* is then opened and the steam enters one of the two chambers *A* and *B*. Sup-

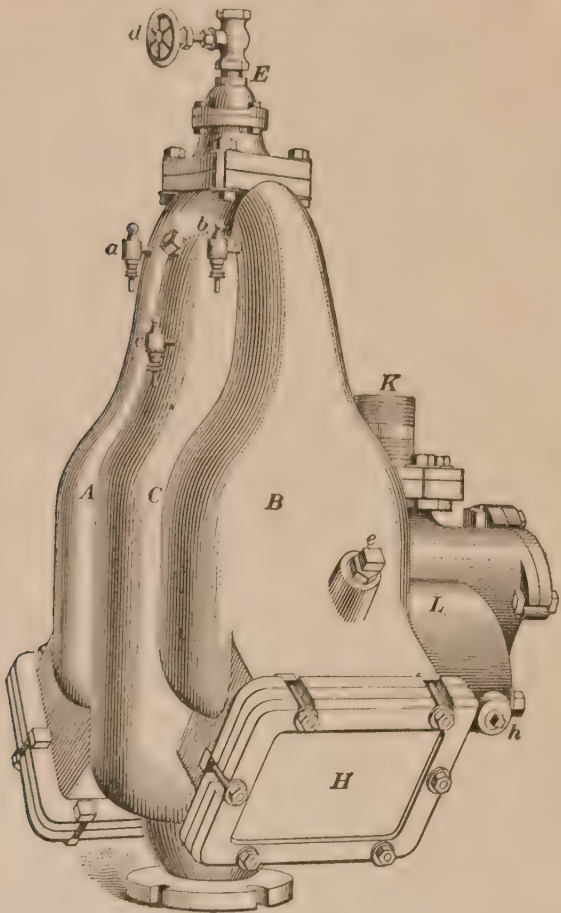


FIG. 33.

pose it enters *B*, the valve *D* being at the right, as shown. The water in *B* will be forced through the delivery valve *N* into and up the column pipe *K*. This will continue until the water level gets below the edge of the discharge

opening P. At this point the steam and water mix in the discharge passage and the steam is condensed, creating a vacuum in B. The pressure in A is now greater than that in B, owing to the vacuum in B, and the ball valve D is shifted to the left, the steam entering the chamber A and

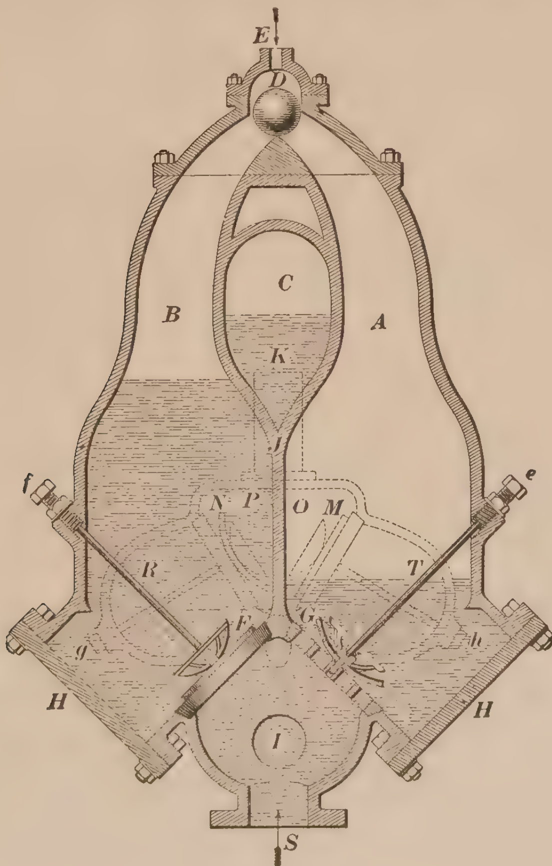


FIG. 34.

driving the water through M into the passage O and column pipe K in the manner just described. While this is being done, the pressure of the atmosphere forces the water up the suction pipe S, opening the suction valve F, and into

the chamber B, filling it. When the suction valve is closed, owing to the reshifting of the ball valve D to the other side, the suction water enters the air chamber C through the inlet I and is brought gradually to rest by the compression of the air in C, thus preventing a shock owing to the sudden stoppage of the inflowing water. When the water in A has reached the level shown, the steam in A is condensed, the ball D is shifted to the right, and B becomes the driving chamber.

85. In Fig. 33 are shown three small air valves a, b, and c. The valve c admits air to the air chamber C, to replenish that which is lost through leakage and through absorption by the water. The valves a and b admit a small quantity of air to the chambers A and B, respectively, just before the suction begins. This injures the suction somewhat, but is necessary for two reasons: First, it acts as a regulator, governing the amount of water admitted to the chambers; and, second, it prevents the steam from condensing before the water gets below the edge of the discharge outlet. These valves open inwards, as before stated. Suppose there is a vacuum in A owing to the condensation of the steam. The atmospheric pressure forces open the valve a and admits a little air to the cylinder. The incoming water compresses this air and soon closes the valve. When the air has been compressed to such an extent as to balance the outside pressure of the atmosphere, the suction valve G will close and no more water can get in. Since the same thing occurs in the other chamber, it is evident that the amount of air admitted controls the amount of water admitted during the suction period, more water entering when there is less air in the chamber and vice versa. The admission of the air is controlled by turning the valves a and b, and these can be so adjusted that the suction valve in either chamber will close at the instant the ball is shifted to the other side, admitting the steam.

Moreover, the air prevents the steam from coming in contact with the water during the forcing process, until the

water level has sunk below the edge of the discharge orifice. Air being a poor conductor of heat, the steam does not condense until the mixture of the steam and water has taken place.

86. When the barometer stands at 30 inches, the pulsometer will raise water by suction to a height of about 26 feet, although it is not advisable to exceed 20 feet, and force it, when necessary, to a height of 100 feet. It has no wearing parts whatever except the valves, which are easily and cheaply repaired. It will work in almost any position, and when once started requires no further attention. There are no parts that can get out of order. It will pump anything, including mud, gravel, etc., that can get past the valves. Its first cost is low and it requires no foundations to set up. There is no exhaust steam to make trouble and no noise.

THE DIRECT-AIR-PRESSURE PUMP.

87. The direct-air-pressure pump here shown is the design of Professor Elmo G. Harris and is one of the simplest forms of pump. The pump is shown in diagrammatic form in Fig. 35. There are two pump tanks *a* and *b*, which are fitted with suction valves *c* and *d* and discharge valves *e* and *f*. The two tanks are connected to the common suction pipe *g* and both discharge into the same discharge pipe *h*. The tops of the pump tanks are connected by pipes *i* and *k* to an air compressor *m*, and by means of an automatically operated four-way cock *l*, either tank can be connected to the compressor side of the air compressor. The operation is as follows: with the cock *l* in the position shown, the tank *b* is connected to the suction side of the air compressor, and hence a vacuum is formed in the tank *b*. Consequently, the water in the supply is forced by atmospheric pressure up the suction pipe *g*, lifts the valve *d*, and passes into the tank *b*. At the same time the tank *a* is connected to the compressor side, and the air pressure on top of the water forces it out, the water holding the suction valve *c*

closed but opening the delivery valve *e* and passing up the discharge pipe *h*. When the tank *a* is nearly empty, the tank *b*

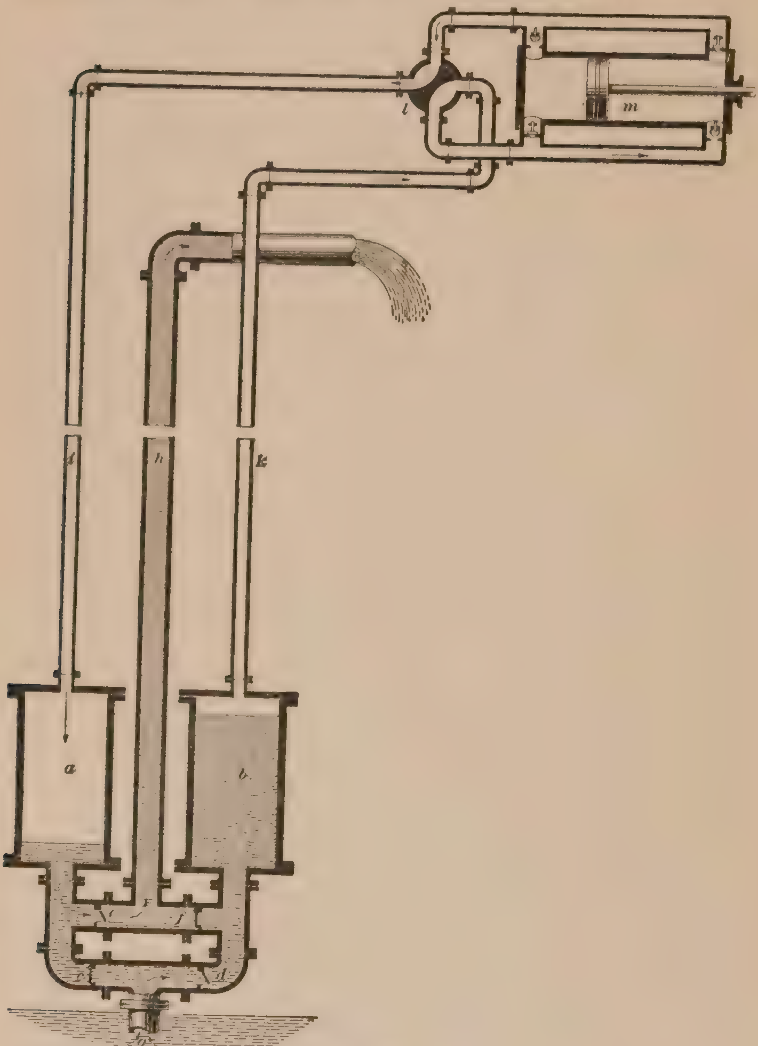


Fig. 35.

is nearly full; the cock *l* is then turned automatically so as to bring the tank *a* in communication with the suction side

of the air compressor and the tank *b* in communication with the compressor side. The water now flows into *a* and out of *b*, and the cycle of operations is repeated as long as the air compressor is working. The height to which water can be forced obviously depends on the pressure to which the air is compressed.

THE POHLÉ AIR LIFT.

88. The **Pohlé air lift** is much used for pumping water from artesian wells; it is operated by means of compressed air and has no moving parts. It is not affected by sand or

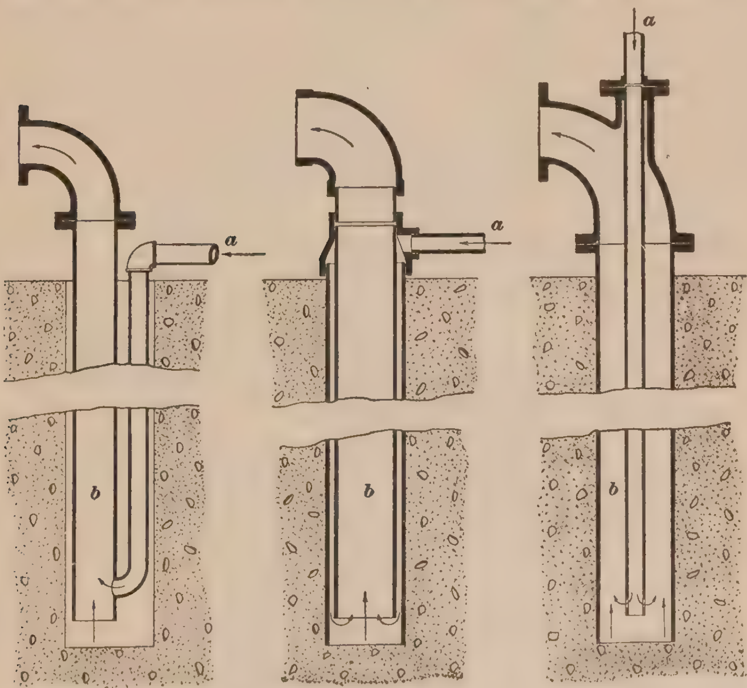


FIG. 36.

grit and the water is benefited to a considerable extent by the action of the air, in that it purifies and cools the water while it is being pumped. Other advantages claimed for

this device is that it increases the yield of an artesian well from two to five times; also, the full area of the well is available for a flow of water. Compressed air is supplied by means of an air compressor at the surface, which may be located in any convenient position, or one air compressor may supply several artesian wells.

89. The operation of the pump is as follows: Two properly proportioned pipes are inserted in the well, using either of the three arrangements shown in Fig. 36. Compressed air is supplied through the pipe *a* to the bottom of the well tube *b*. At the beginning of the operation the water inside and outside of the pipe is at the same level. When air is forced in through the pipe *a*, it forms alternate layers with the water, so that the pressure per square inch of the column thus made up of air and water inside of the water pipe is less than the pressure per square inch outside the pipe. This difference of pressure causes a continuous flow from the outside to the inside of the water pipe, and its ascent is constant and is free from shock or noise of any kind. The strata of compressed air in their ascent prevent any slipping back of water. As each stratum progresses upwards to the spout, it expands on its way in proportion to the overlying weight of water, so that the pressure of the air gradually becomes less and finally reaches the atmospheric pressure.

WATER ENDS OF RECIPROCATING PUMPS.

TYPES OF WATER ENDS.

90. Reciprocating pumps are either single-acting or double-acting. Single-acting pumps are either lift pumps, one of which is shown in Fig. 25, or outside-packed plunger pumps with one plunger, as shown in Fig 26, or outside-packed double-plunger pumps, as shown in Figs. 29, 30, and 32. Double-acting pumps are force pumps of the piston

or plunger pattern. Piston pumps, by reason of their construction, are inside-packed, and such a pump is shown in Fig. 6. Double-acting plunger pumps are inside-packed or center-packed. Attention is here called to the fact that outside-packed double-plunger pumps are often, but erroneously, considered as double-acting. While they give a discharge equal to that of a double-acting plunger pump, it is obtained by combining two single-acting plunger pumps to discharge into the same delivery pipe, and hence it is incorrect to call such a pump a double-acting pump. They are properly called **duplex pumps.**

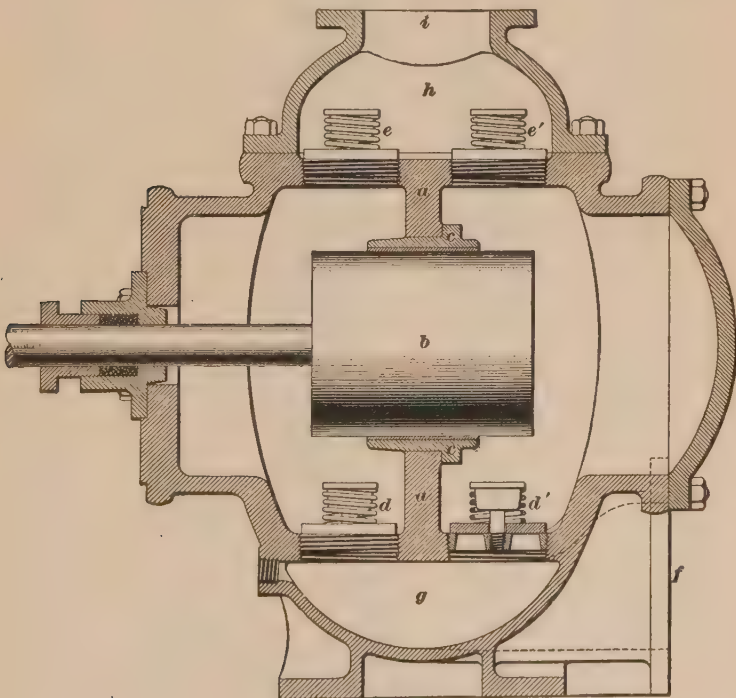


FIG. 37.

91. Fig. 37 shows the water end of a **double-acting inside-packed plunger pump.** The pump chamber is divided into two parts by a partition a, through which the

plunger *b* works back and forth. A water-tight joint between the plunger and partition is made either by a closely fitting

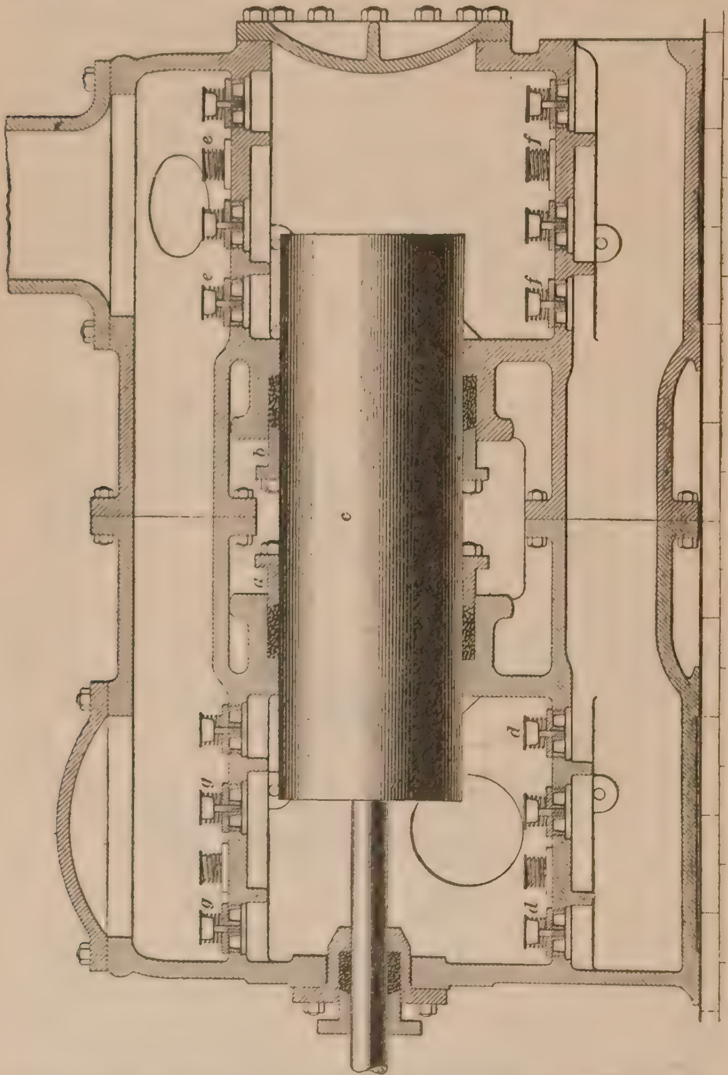


FIG. 38

bronze-lined bushing *c* or a regular stuffingbox and gland and fibrous packing. On either side of the partition is a set

of suction valves *d*, *d′* and delivery valves *e*, *e′*. The water enters the pump through the suction pipe, which is connected at *f* and flows into the suction-valve chamber *g*, from whence it passes to either side of the partition *a* and then into the delivery-valve chamber *h* and into the delivery pipe connected at *i*. When the plunger moves to the right, it displaces the water on the right of the partition *a*; the suction valve *d′* is closed by the pressure existing there, while the delivery valve *e′* is open and the water discharges into *h*. At the same time the plunger creates a partial vacuum at the left of the partition *a* and, hence, water flows through the open suction valve *d* into the left pump chamber. The delivery valve *e* is kept closed by the pressure in *h*. When the plunger moves to the left, the suction valve *d′* and delivery valve *e* open and the suction valve *d* and delivery valve *e′* close. It is thus seen that the pump discharges during either stroke of the plunger, i. e., the pump is double-acting.

92. Fig. 38 shows a sectional view of the water end of a center-packed double-acting plunger pump, the stuffing-boxes *a* and *b* being used for packing the plunger *c*. The action of the pump is identical with that of the pump shown in Fig. 37, that is, when the plunger moves to the right the suction valves *d*, *d* and delivery valves *e*, *e* are open and the suction valves *f*, *f* and delivery valves *g*, *g* are closed. When the plunger moves to the left, the suction valves *f*, *f* and delivery valves *g*, *g* are open and the suction valves *d*, *d* and delivery valves *e*, *e* are closed.

93. The water end of a double-plunger pump for high pressures is shown in Fig. 39. The two plungers *a*, *b*, as usual, are connected by yokes and side rods outside of the pump. The rods *i*, *i* tie the water end to the steam end. Each plunger has its own suction valve *e* and delivery valve *f*. The suction valves communicate with a common suction chamber, to which the suction pipe *c* is attached. At *d* the discharge pipe is shown. Plugs *g*, *g* when removed give access to the valves. A standard *h* supports the water end

on its foundation. The illustration clearly shows that each plunger is single-acting, but that the discharge is equal to that of a double-acting pump. Pressure pumps do not differ

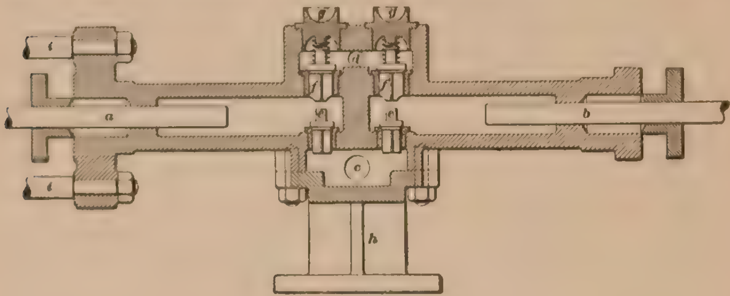


FIG. 39.

in their operation from ordinary pumps; all parts are simply made extra heavy so as to stand the high pressure, and for very high pressures steel is substituted for cast iron in the water end.

94. Fig. 40 shows in diagrammatic form two forms of a plunger pump that is double-acting and is known as a **differential pump.** Its distinguishing feature is that it needs only one set of suction valves and delivery valves. Fig. 40 (*a*) shows the arrangement used for two plungers *a* and *b*, which are connected together by yokes and side rods. In Fig. 40 (*b*), the two plungers are connected directly together. In both designs one plunger, as *a*, has exactly double the area of the other plunger *b*. This fact must be carefully borne in mind. Since the stroke of both plungers is the same, it follows that the larger plunger in Fig. 40 (*a*) will displace double the quantity of water that the smaller plunger displaces. In Fig. 40 (*b*), the left-hand side of the plunger *a* displaces double the quantity of water displaced by the plunger *b*. In both designs *c* is the suction valve and *d* the delivery valve.

95. The operation of the differential pump shown in Fig. 40 (*a*) is as follows: The pump being filled with water and the plungers moving to the right, the suction valve is

open and the delivery valve closed. The plunger *b*, or the right-hand side of the plunger *a* in Fig. 40 (*b*), forces a volume of water equal to its displacement out of the chamber *e* and up the delivery pipe *f*. At the same time, double the volume of water is drawn into the suction chamber *h*.

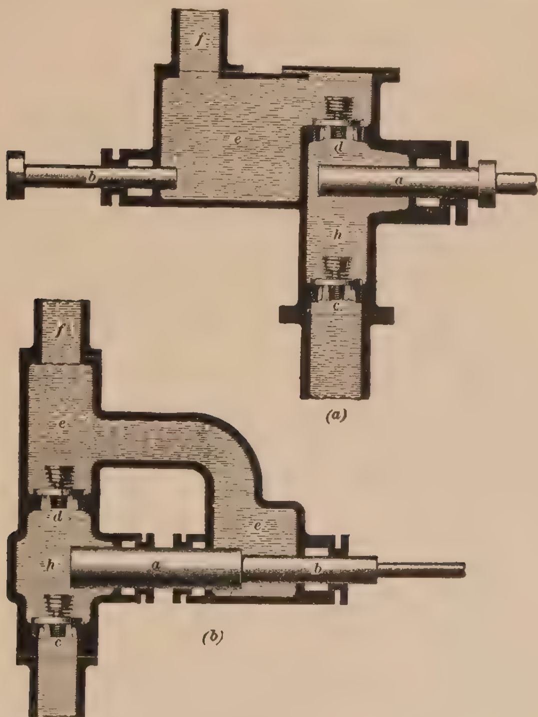


FIG. 40.

Now, assume that the plungers move to the left. The suction valve is then closed and the delivery valve is open, and double the quantity of water discharged during the stroke to the right now flows into the chamber *e*. But while this is going on, the volume of the chamber *e* increases by the receding of the plunger *b*, or the outward movement of the plunger *a* in Fig. 40 (*b*), by an amount that at the end of the stroke is equal to exactly one-half the amount discharged into it, so that the outflow into the delivery pipe is

only one-half of that discharged into the chamber *c*. This outflow is equal to the displacement of the small plunger, or the right-hand end of the plunger *a* in Fig. 40 (*b*), and hence the same amount of water is discharged during both strokes.

RIEDLER PUMPS.

96. Development.—Riedler pumps are the invention of Professor Riedler and are a type of pump designed for running at very high speeds. By study, experimenting, and careful noting of cause and effect, he discovered several very important phenomena. He found that there was much greater resistance to the flow of water through the valve passages and ordinary pumps than was before this thought to exist. He further found that the slip of ordinary valves is very large, and that even when small has a great tendency to cause severe hydraulic shocks throughout the pressure parts of the pump. He also was aware that the frictional resistance to the passage of a certain quantity of water through a large number of small openings is much greater than that existing when the same quantity of water passes through a single opening equal to the combined area of the smaller ones. With these facts in view, Professor Riedler designed a pump valve having the useful valve area as large as possible and containing as few separate passages as is consistent with good construction. He substituted one large valve for many small ones, thus decreasing the friction of the water in the valve passages. The reduction of the slip was accomplished by arranging a mechanical controlling device, whereby at the proper time and without restricting the water passage the valve was closed. The mechanical controlling device further assists in the reduction of friction in the valve passages, as it permits the valve lift to be high, thus increasing the effective area.

97. The first pumps fitted with Riedler valves were constructed in 1884, since which time more than 1,500 pumps have been built. These pumps are adapted to any service

to which pumping machinery may be applied. They are built in all sizes, ranging in capacity from 115,000 gallons in 24 hours to 20,000,000 gallons in 24 hours, and are working under heads as high as 2,480 feet and at speeds as high as 120 revolutions per minute, and with piston speeds as high as 606 feet per minute, which, by the way, is the average speed of steam pistons.

98. Valve Gear.—Fig. 41 shows an outside view of a direct-connected, electrically driven, differential Riedler pump having the plunger arrangement shown in Fig. 40 (*b*). The pump valves are closed by cranks, the crank *a* operating the suction valve and the crank *b* the delivery valve. The two cranks are operated from a wristplate *c* similar to that of a Corliss engine and to which they are connected by the rods shown. The wristplate is rocked back and forth by the eccentric *d* on the crank-shaft, to which it is connected by the eccentric rod *e*. The plungers are driven by a crank, as shown.

99. Riedler Valve.—Fig. 42 shows a detail of the improved Riedler suction and delivery valve. Both suction and delivery valves are alike in these pumps except as regards the flange for securing them to the pump chambers. The valve proper consists of three concentric bronze rings *a*, *b*, and *c*, each of which is cast in one piece and which are set into a spider *d* having eight arms. This spider is free to move up and down on the central valve post, or valve spindle *e*. This valve rests on a heavy cast-steel valve seat *f* having three annular openings *a'*, *b'*, and *c'*. The valves proper are not rigidly connected to the spider, but each valve is free to form its seat with the valve seat and independent of the spider or each other. A leather ring between the valve proper and the spider serves to make an absolutely tight joint. A circular nut *g* is secured to the top of the hub of the valve spider *d* and holds in place a steel pressure plate *h*. This pressure plate rests on top of a spring cap *i*, below which a spring *k* of soft rubber is placed. This rubber allows of a certain amount of yield between the valves

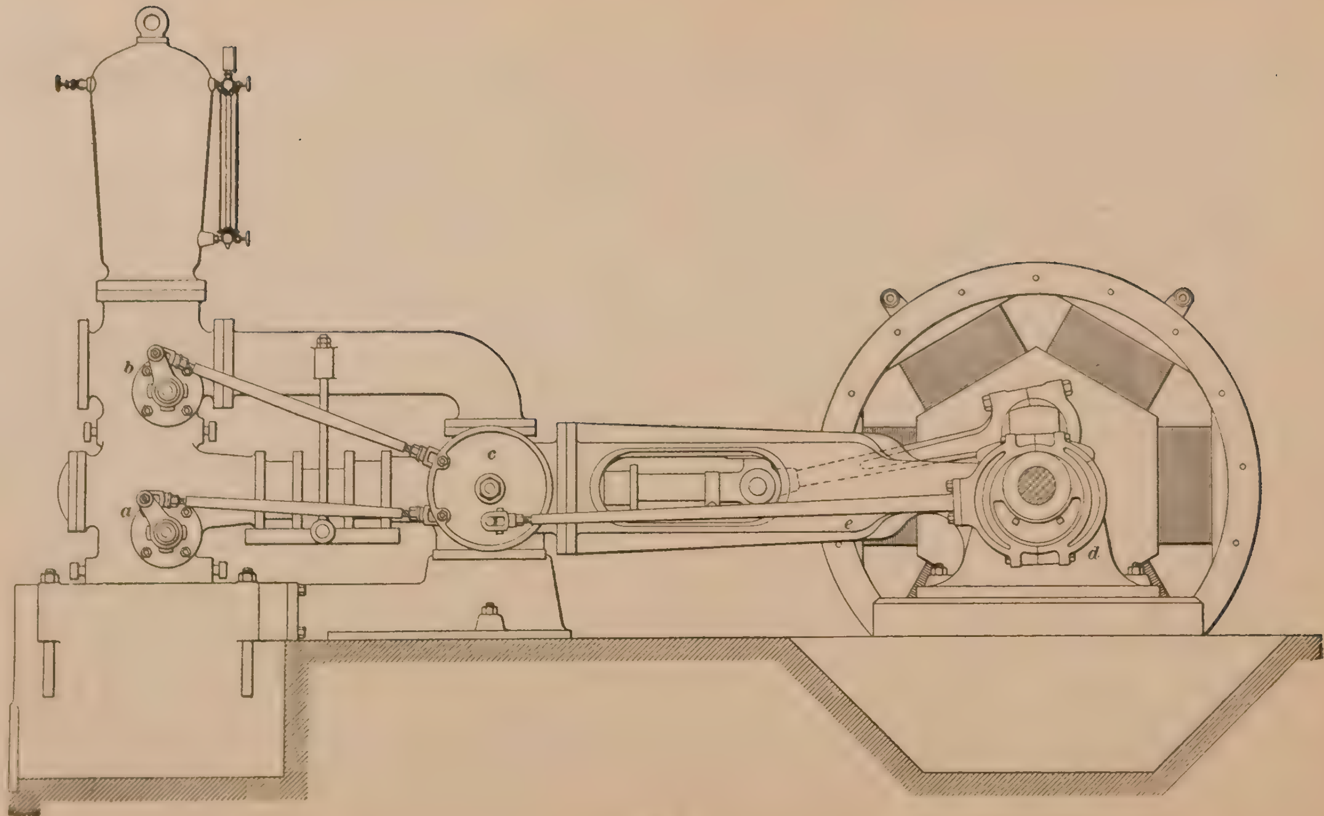


FIG. 41.

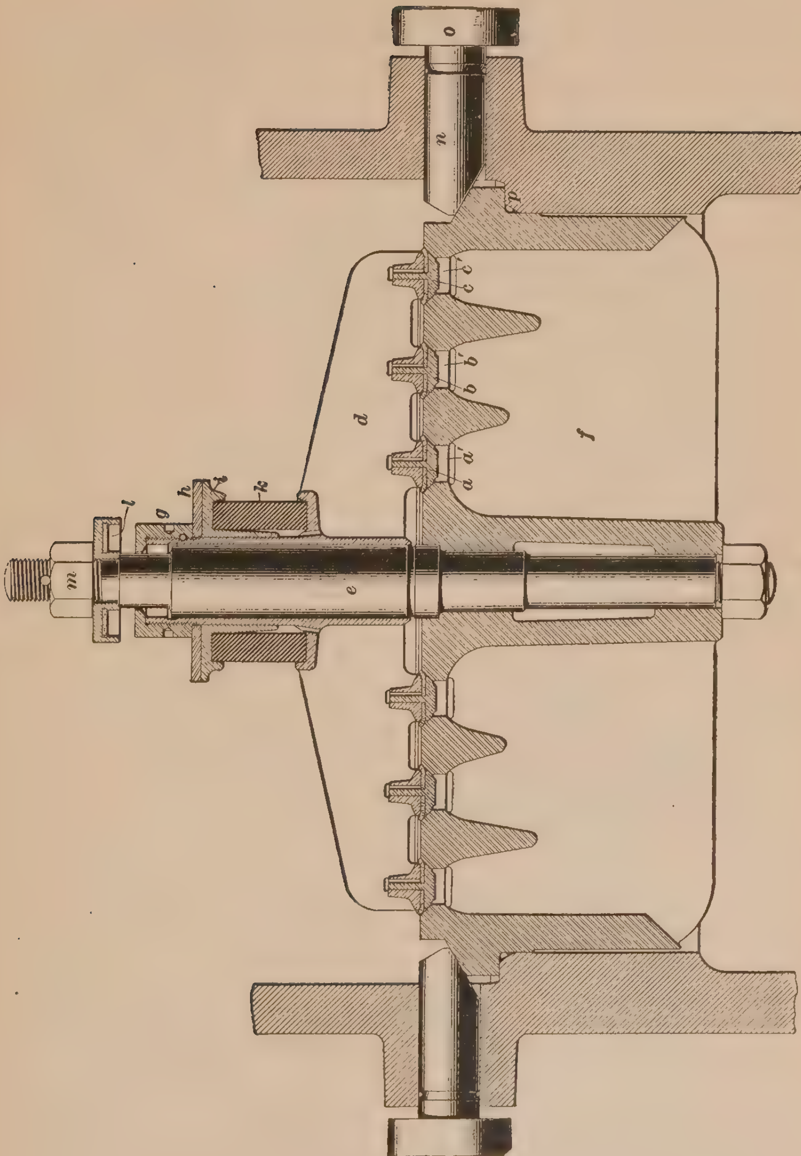


Fig. 42.

and its seat in case any foreign matter should get between them. Two steel fingers, not shown in the drawing, press upon the pressure plate and serve to close the valve just before the piston reaches the end of its stroke. A water cushion *l*, the object of which is to prevent the valve from striking its stop when opening, is secured to the top of the spindle *e* by the nut *m*. The nut *g* is closely fitted to the chamber in *l* and traps the water in front of it, thus making a hydraulic cushion. The valve seats are secured in the valve chambers by wedge-shaped plugs *n*, *n*, which are forced in by studs and nuts through the gland *o*, the effect being to force the valve seat *f* hard down on its bearing *p* in the pump chamber.

100. Fig. 43 is a perspective view of the Riedler valve and seat, showing the operating mechanism by means of which the valve is seated. All visible parts are lettered the

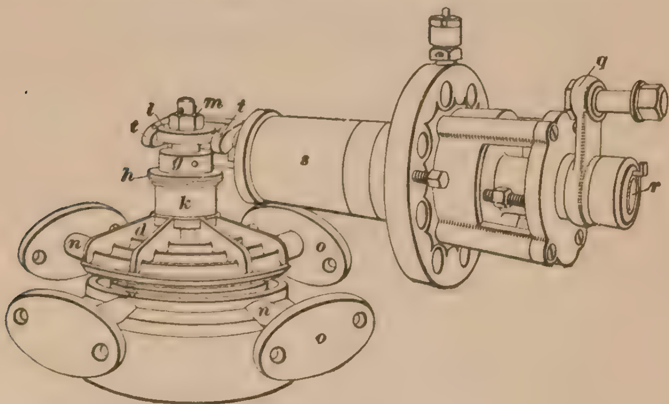


FIG. 43.

same as in Fig. 42. The crank *q* is operated from the wristplate shown in Fig. 41; it is keyed to a shaft *r*, which passes through a stuffingbox *s* bolted to the valve chamber and carries a forked crank at its inner end. The jaws or fingers *t*, *t* of the forked crank press upon the pressure plate *h* to seat the valve at the proper time. The motion of the fingers is so timed in relation to the motion of the

plungers that the fingers are clear of the pressure plate *h* when the plungers begin to deliver water, thus leaving the valve free to open.

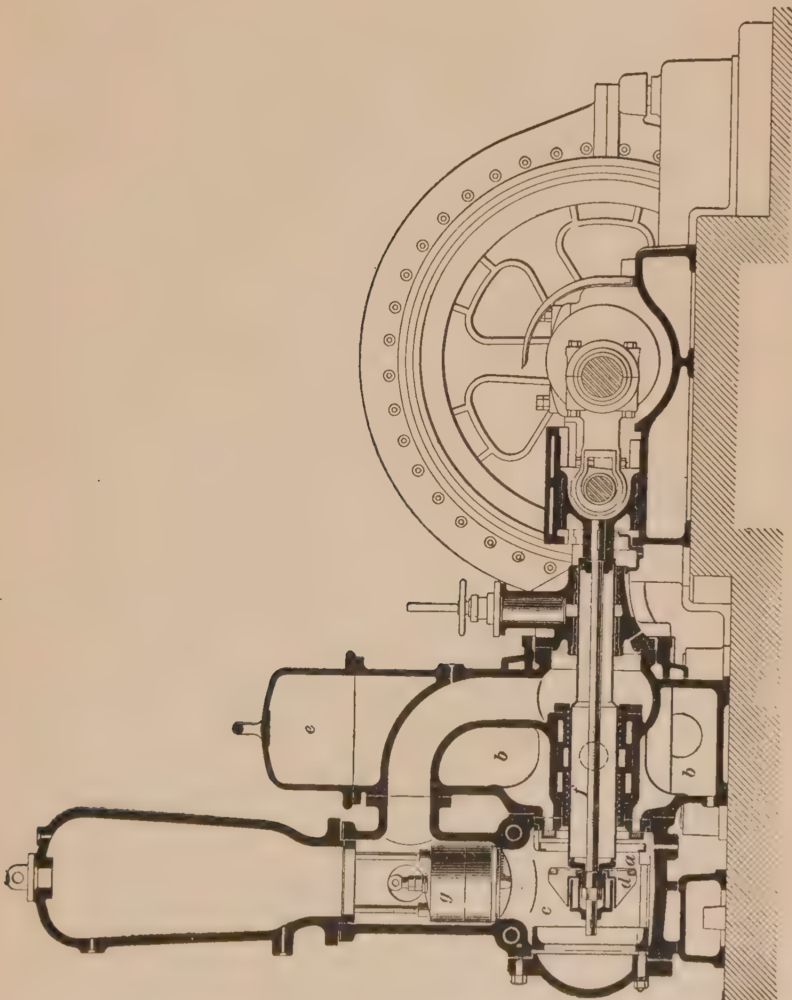


FIG. 44.

101. The Riedler valve is by no means confined only to water pumps. It has been and is used successfully for high-pressure air and gas compressors. The Riedler pump may

be driven by a steam engine, electric motor, turbine water-wheel, by belting, or in any other convenient manner.

102. Riedler Express Pump.—A type of Riedler pump that has recently been brought out for running at a very high speed is called the **Riedler express pump** and is shown in Fig. 44. Although the ordinary Riedler pump can be run at speeds as high as 150 revolutions per minute and sometimes faster, conditions arise requiring a much higher speed, and to meet this condition this special design, which may be run at speeds as high as 300 revolutions per minute, has been developed by Professor Riedler. The main feature of this pump—in fact, the part that permits running at such high speeds, is its suction valve. As will be seen by referring to the figure, the suction valve *a* is annular in form and is concentric with the plunger; it lifts in the direction opposite to that of the plunger when on its suction stroke, the water flowing from the suction chamber *b b* into the valve chamber *c*. At the end of the suction stroke a buffer *d* mounted upon the end of the plunger drives the suction valve to its seat, making it certain that the valve is seated when the plunger starts on its delivery stroke and allowing practically no slip. A high suction air chamber *e*, containing a column of water, is placed above the suction valve, making it certain that the pump will fill as the plunger *f* makes its suction stroke. The delivery valve is shown at *g*. It will be noticed that this pump is of the differential type.

103. The chief point of advantage of the express pump is that it may be connected to high-speed motors. It is of small dimensions compared to the quantity of water it can handle, and thus consequently low in first cost. About thirty of these pumps have been constructed up to the year 1901, ranging in capacity from 152,000 gallons in 24 hours to 7,600,000 gallons in 24 hours, and in speed as high as 300 revolutions per minute, pumping against a head of 820 feet; others have been built to pump against a head of 1,800 feet at 200 revolutions per minute.

PUMPS.

(PART 2.)

DETAILS OF PUMP WATER ENDS.

PUMP PLUNGERS.

CONSTRUCTION.

1. The smaller sizes of pump plungers are usually made of solid round bars of metal turned smooth, so as to work through a stuffingbox with as little friction and wear as possible. For larger sizes the plungers are frequently of cast iron and are often made hollow to reduce the weight and amount of material required. Incidentally, it may be remarked that a hollow plunger is easier to move than a solid one, all other conditions being equal. This is due to the fact that the water buoys up a hollow plunger more than a solid one. In large horizontal pumps hollow plungers are often so proportioned that they actually float in the water, thus relieving the stuffingboxes of the weight of the plungers and reducing the wear.

2. Fig. 1 shows a simple form of solid plunger pump, such as is often used for feeding boilers. The plunger works through a stuffingbox of the ordinary pattern, packed with hemp or some of the common types of soft piston-rod packing.

3. Fig. 2 shows three styles of large, hollow, cast-iron

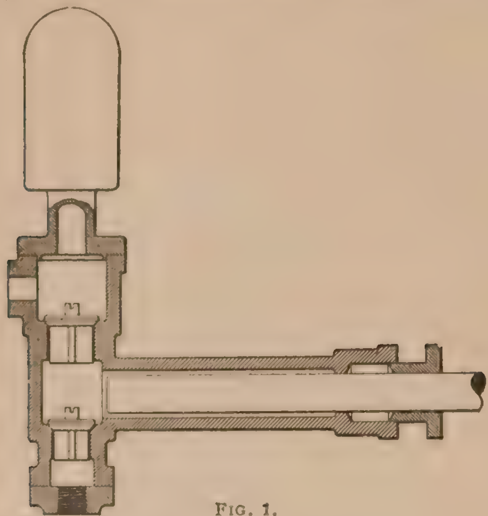


FIG. 1.

plungers, with methods of attaching them to the pump rods. The packing for these plungers, when used for moderate pressures, is usually hemp contained in a stuffingbox of the ordinary pattern.

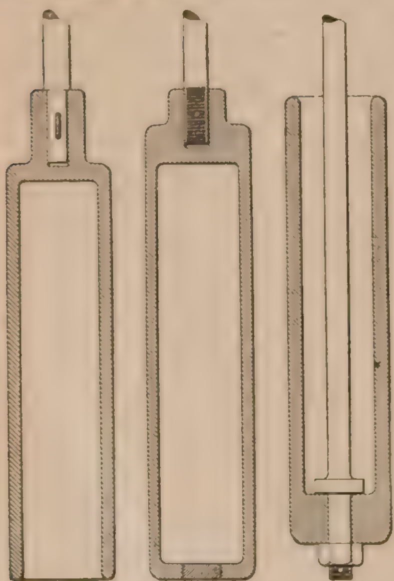


FIG. 2.

PLUNGER PACKING.

4. When the pressure under which the pump works is very heavy, **U**-shaped leather packing is sometimes used. Fig. 3 shows three methods of holding these **cup leathers,** as they are called. The section at (*b*) shows the leather *o* held in a recess cast in the upper end of the

pump cylinder. In this case it is necessary to remove the plunger *D* in order to insert a new leather or to examine an old one. Experience also shows that the leather bears against the plunger with the greatest force at the bend *B* and fails at that point first. In (*c*) the leather is held in its recess by a gland *s*, and is also supported by a brass ring *C*,

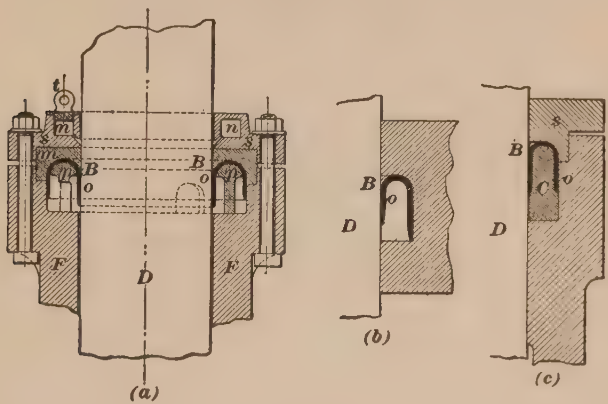


FIG. 3.

which prevents the severe pressure of the leather against the plunger at *B*. A more elaborate packing is shown at (*a*); the gland *s* is lined with a brass ring *m*, which holds the leather *o* down on a brass supporting ring *p*. A chamber *n* in the gland serves to hold oil for lubricating the plunger.

The form of packing shown at (*b*) is cheap, but in addition to the difficulty of inserting the leather, it is difficult to cast the recess so that it will fit the leather properly. In either of the forms shown in (*a*) and (*c*), the gland can be accurately turned to bear against the curved portion of the leather, thus forming a better support and increasing the life of the packing.

5. Fig. 4 shows an inside-packed plunger with a removable stuffingbox designed for hemp packing. This construction is better than merely providing a close-fitting bushing, especially when the water is gritty and thus liable to wear the plunger.

Inside-packed plunger pumps have several disadvantages. When the packing becomes worn, the heads of the pump cylinder must be removed in order to tighten or renew it, and,

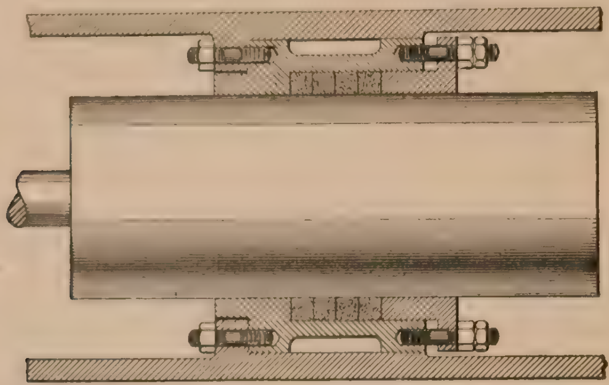


FIG. 4.

besides, there is no way of detecting leakage when the pump is working. With gritty water, especially when working under high pressures, these disadvantages become serious.

6. Fig. 5 shows a good arrangement of plunger, stuffingbox, and gland. This type of plunger and stuffingbox is

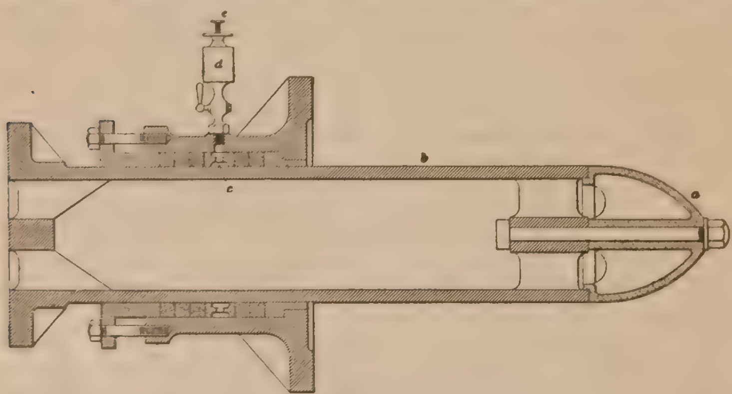


FIG. 5.

much used in mining pumps. The plunger cap *a* is made of acid-resisting metal, while the plunger *b* proper is made of cast iron, it having been found in mining work that the

plunger cap or point is the only part that is attacked by acid water. Apparently the play of the plunger through the stuffingbox and grease prevents the water attacking its surface. An improved form of **grease ring** is shown at *c*. This ring fits into the stuffingbox and is placed between the rings of fibrous packing. It is recessed both inside and outside and has several holes by which the outside recesses connect with the inside recesses. The outside recess is in connection with the grease cup *d*, which is provided with a cock. When it is desired to grease the plunger, the cock is opened and the grease forced in the space around the grease ring by the screw *e* on top of the grease cup. This is done once or twice during the day, and the cock is then closed so as to relieve the grease cup of the water pressure and to prevent consequent leakage. The stuffingbox is bolted directly to the pump chamber, which may be of any type, but for high-pressure mine work it is generally circular. This type of plunger and stuffingbox has been used with much success in the anthracite coal regions.

PUMP PISTONS.

7. Pistons for force pumps are made in a variety of forms. Fig. 6 shows a piston with fibrous packing held in place by a follower. The follower is fastened to the piston by means of an extension of the piston rod beyond the nut that holds the piston in place.

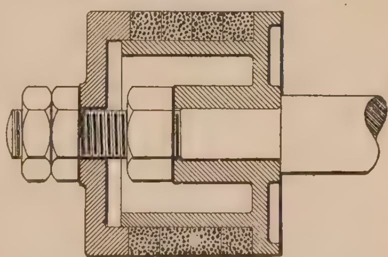


FIG. 6.

8. An excellent packing for small pistons is shown in Fig. 7. It consists of a metallic piston made up in three parts, between which are clamped two cup leathers, as shown.

9. Pistons for suction and lift pumps must be provided with valves that allow free passage for the water through

the piston in one direction and prevent its return. These

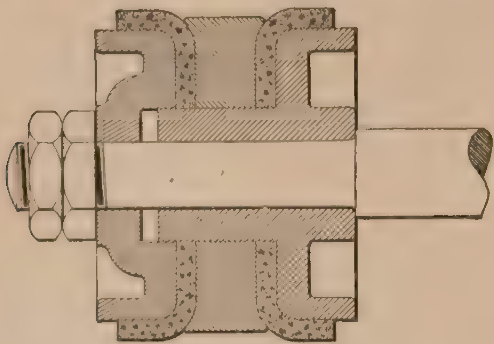


Fig. 7.

valves may be of any design that will furnish the required area of passage and at the same time will be strong enough to withstand the pressure of the water.

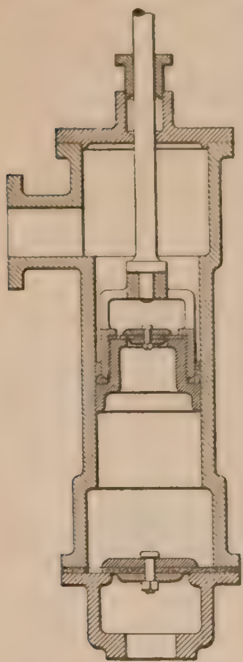


Fig. 8.

10. For small pumps and moderate lifts, leather **clack valves,** Fig. 8, are often used. They consist simply of a leather disk held at one side and strengthened by a metal plate on top. The leather when wet forms an excellent hinge and a tight valve. Leather clack valves are also used for the suction and delivery.

11. For lift pumps working under high pressures, the valves shown in Fig. 9 give good results. The piston shown at (*a*) has a rubber disk valve working on a gridiron seat. The valve is guided by a central spindle *s* and is held on its seat by a light helical spring that acts on a plate on top of the rubber disk. This piston is very long and has no separate packing.

12. The valve shown at (*b*) is for very heavy pressures. It consists of a metal disk guided by a central spindle *s* and held down by a helical spring in the same manner as the

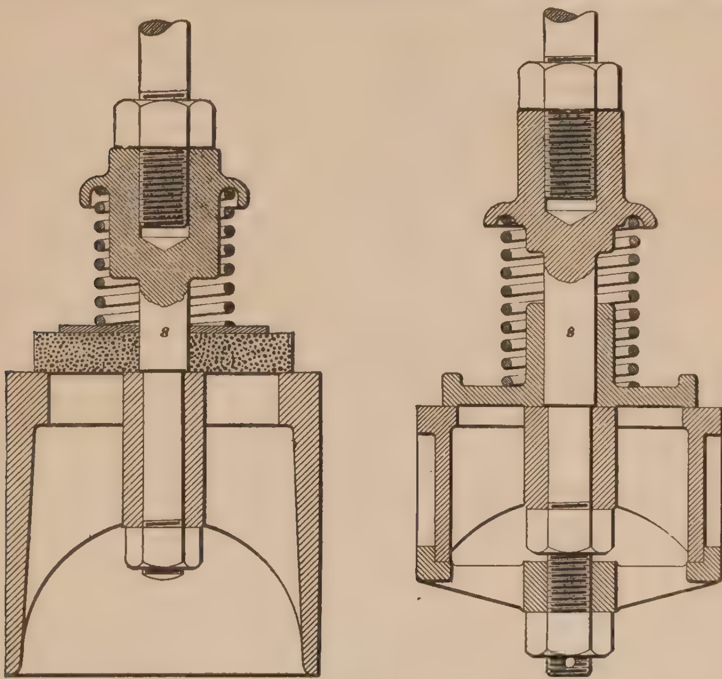


FIG. 9.

rubber valve. The piston is made with a follower plate for the purpose of holding a fibrous packing in the same manner as the piston shown in Fig. 6.

PUMP VALVES.

REQUIREMENTS.

13. The most important details of a pump of any kind are the valves. They must be so designed and constructed that they will fulfil all the following conditions as thoroughly as possible:

(*a*) They must open freely under a light pressure.

(*b*) The net area of the passages through the valves should be great enough to limit the velocity of flow through them to 240 feet per minute.

(*c*) The lift of the valves should be small.

(*d*) The passages for the water should be as direct as possible.

(*e*) The valves must close tightly under all conditions.

(*f*) The valves and their seats must be durable and of such materials as are not easily affected by the impurities in the water.

(*g*) The valves must return to their seats quickly and without shock as soon as the current through them is stopped.

(*h*) The valves and seats must be easily repaired or removed when worn.

A great variety of valves have been designed with a view of satisfying these requirements, taking into consideration the widely varying conditions under which pumps must work.

CONSTRUCTION.

14. Disk Valves.—Fig. 10 shows two valves of a type much used in all classes of pumps for ordinary pressures and service. The valve *v* consists of a vulcanized India-rubber disk that rests on a gun-metal or brass seat *s*. The seat is threaded at *t*, so that it can be screwed into the deck of the valve chamber and thus can be easily removed. The part of the pump chamber

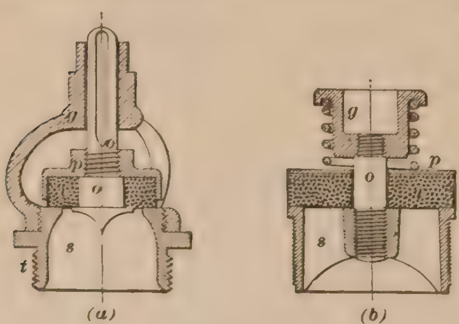


FIG. 10.

that contains the valves is usually called the **valve deck**,

and it is spoken of as the **suction valve deck** and **delivery valve deck** in accordance with the kind of valves it carries. In the design shown at (*a*), the valve is fastened to a spindle *o* by a cap *p*. The spindle is guided by a cage-shaped guard *g* screwed on to the valve seat. The lower end of the spindle is made conical, so as to change the direction of motion of the water gradually and to reduce the resistance to flow. In the design shown at (*b*), the spindle *o* is screwed into the valve seat and carries a guard *g*. A helical spring between this guard and the plate *p* helps to seat the valve quickly.

The size of these valves varies from 2 to 6 inches in diameter, the most common size for ordinary conditions being 3 inches.

FIG. 11.

15. When used for pumping *hot* water, the disk must be made of a composition that will not be affected by the heat and for very high pressures metal disks are used, generally of the form shown in Fig. 11.

16. Fig. 12 shows the construction of a large disk valve, such as is often used in mine pumps. The valve seat *A* is held in place by the flange *B* and is perforated, as shown in the top view of the seat, by a large number of small holes. The valve *C* is made of soft rubber and is placed within the bronze or composition cap *D*. The head of the bolt *E* forms a stop and the spring *S* assists the valve in closing.

17. Clack Valves.—A section of a clack valve is shown in Fig. 13. The **clacks** *A* and *B* are lined with leather on the bottom so as to make a tight fit on the seat without having to do much fitting. A stop *C* prevents the valves opening too far, while *E* is the pin on which the clacks are hinged. A cylindrical casing *D* forms the valve seat; it may be easily renewed when worn. These valves are of the type known as the **butterfly valve,** and are much used for pit pumps at mines on account of their cheapness and simplicity of construction.

18. Single-Seat and Double-Seat Valves.—A **single-seat valve** that is suitable for high pressures, up to heads of 500 feet, is shown in Fig. 14, where A is the valve; B is

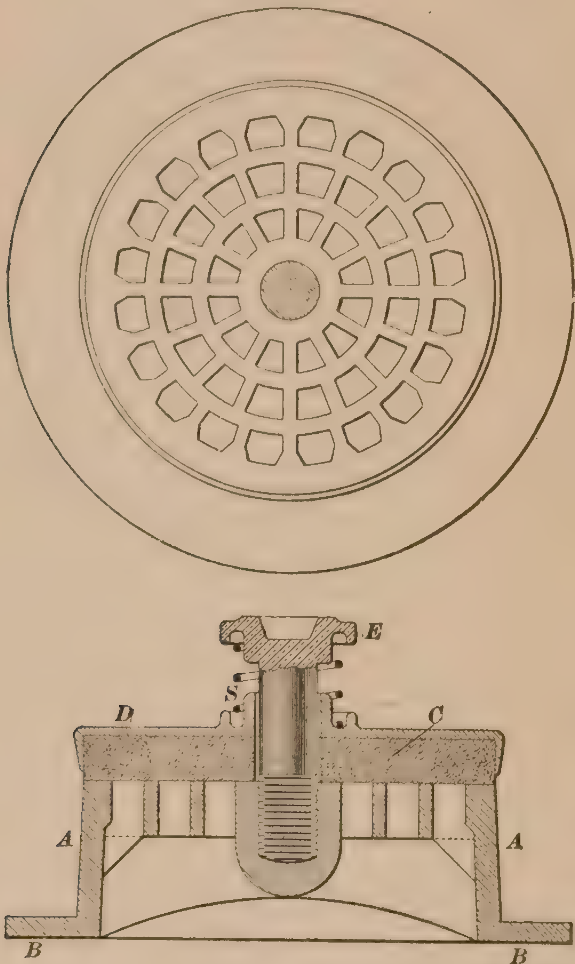


FIG. 12.

a stem solid with the valve that acts as a guide inside the bearing D; and C, C, C, C are rubber rings which are kept in position by means of the stem and are separated by the

washers *E*, *E*, *E*. These rings prevent shock as the valve lifts and also help to close it quickly, thus serving the same purpose as the helical spring in Fig. 10 (*b*).

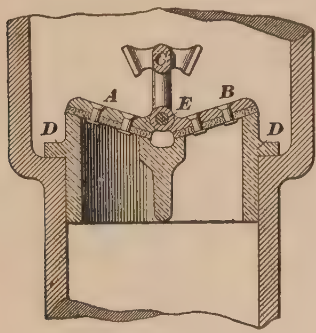


FIG. 13.

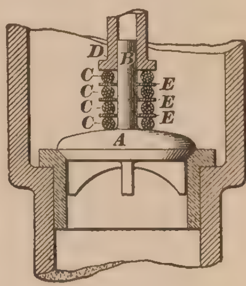


FIG. 14.

19. A section of a **Cornish double-seat valve** is shown in Fig. 15. This valve gives excellent results when used in

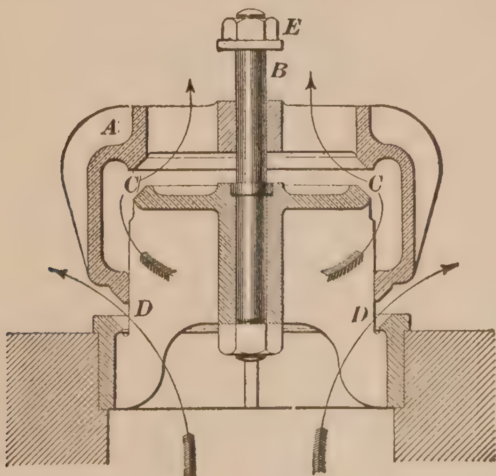
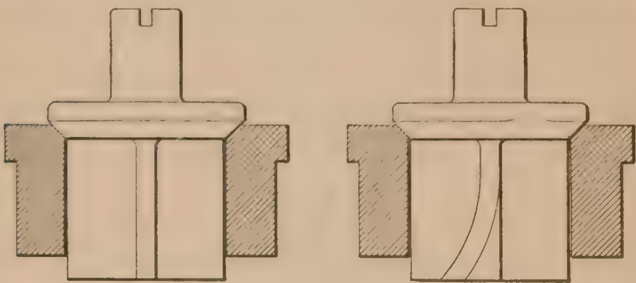


FIG. 15.

large pumps working under high pressures and has been applied to pumps working under heads up to 700 feet. It is called a *double-seat* valve because it has two seats and two

openings for discharge. The casing *A* slides on the vertical stem *B*, its lift being regulated by the nut and washer *E*; when down, it rests on the valve seats *C* and *D*. When the pressure below becomes greater than that above, it raises the casing, and the water is discharged through the circular openings at *C* and *D*. The rib around the outside of the casing is for the purpose of strengthening it. The valve seats are conical. The figure shows that one opening discharges the water under the lower edge of the valve and the other through the inside.

20. Wing Valves.—The **wing valve** shown in Fig. 16 (*a*) is largely used in power pumps for feeding boilers and in



(*a*) (*b*)

FIG. 16.

hydraulic pumps for high pressures. The valve and seat are made either of hard brass or of gun metal and are ground together to secure tight closing. The lower portion of the wings is sometimes curved as shown at (*b*), the object being to give the valve a partial rotation at each stroke of the pump. This compels it to seat at a new place with each stroke and tends to wear the valve and seat more evenly.

21. Pot Valves.—Fig. 17 (*a*) is a sectional view of a **pot valve.** This type of valve is used principally on mining pumps for lifts up to 1,000 feet. They are made separate from the pump chambers and may be readily replaced when broken or worn. The cover *a* is secured by hinged bolts, so that it may be quickly removed for access to the valve *b* and the valve seat *c*, which is made of composition and pinched

between the pot and the pump chambers. The valve spring *d* surrounds the valve guide *e*.

22. Fig. 17 (*b*) shows a type of pot valve used for high lifts up to 1,200 feet. The valves are made small and faced with hard rubber; a group of them is placed in one heavy

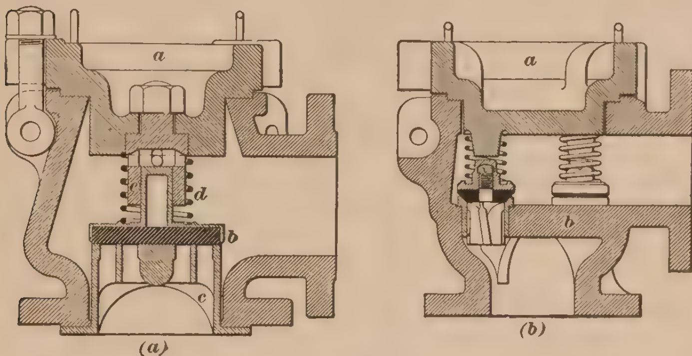


FIG. 17.

pot which is bolted to the pump chamber. Access to the valves may be had by removing the cover *a*. The valve seats are made of composition bushings forced into the valve deck *b*.

AIR CHAMBERS.

PURPOSE.

23. Even in double-acting pumps there is an interruption of the flow at the end of the stroke, when the piston changes its direction of motion. This has the effect of bringing the column of water in the suction and discharge pipes to rest at the end of each stroke, and this column of water must be set in motion again as the next stroke is made. If the pipes are long, the force required to stop and start the water will be very great, and there will be a severe shock at the end of every stroke that will absorb power and subject the pump and pipes to great stresses.

This difficulty is removed and the flow through the pipes is made more continuous and steady by the use of **air chambers.** An air chamber is a vessel containing air and is attached either to the pump just outside of the discharge valves or to the discharge pipe near the pump. While small duplex pumps are often run without an air chamber, it is better in general to fit one to all pumps, since its effect will always be beneficial.

DELIVERY AIR CHAMBERS.

24. Principle of Action.—Fig. 18, which shows an air chamber attached to the discharge pipe of a single-acting

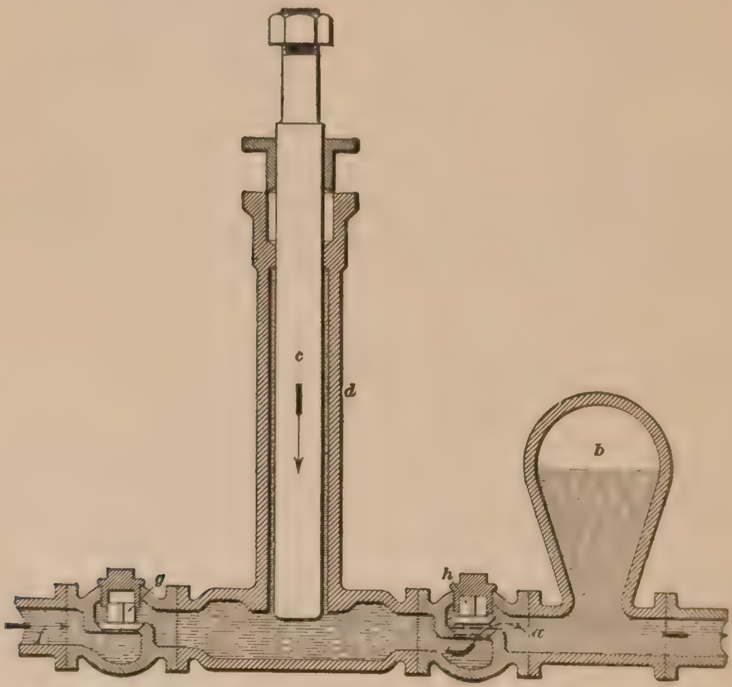


FIG. 18.

plunger pump *d* for boiler feeding, will illustrate the principle of action of an air chamber. The water, after being

drawn in through the pipe *f* past the valve *g*, is forced by the plunger *c* past the valve *h* into the discharge pipe *a*, part of it flowing into the air chamber *b* and compressing the air therein. When the plunger reaches the end of its stroke and no more water is being forced into the discharge pipe, the compressed air in the air chamber forces the extra water out through the discharge pipe. In this way the air chamber acts as a *reservoir* that receives its supply during the inward motion of the plunger and gives it out again in a nearly steady stream. The air in the air chamber acts as a spring that absorbs the extra force during the inward stroke of the plunger and gives it out during the return stroke, thus relieving the pump and pipe of shocks and providing a nearly constant rate of flow from the discharge.

25. Size of Delivery Air Chamber.—The proper size of an air chamber depends on the type of pump, the speed at which it works, the length of the discharge pipe, and the pressure head against which the pump works. For ordinary double-acting pumps working against moderate pressures and at ordinary speeds, the cubical contents of the air chamber should be not less than 3 times the piston displacement. For pressures of 100 pounds per square inch and upwards or for high piston speeds (as in the case of fire pumps), the capacity of the air chamber should be at least 6 times the volume of the piston displacement for a single stroke.

26. Loss of Air.—Under the increased pressure in the air chamber, the air is absorbed by the water and gradually passes off with it. In this way all the air will finally pass off and the chamber will be made useless if no means are provided for renewing the supply.

27. A simple device for maintaining the supply of air in the air chamber of large pumps is shown in Fig. 19. A piece of $2\frac{1}{2}$-inch wrought-iron pipe *c* about 30 inches long is connected to the end of the pump cylinder *a* in a vertical

position, by means of a gate valve *b*, or cock. A $2\frac{1}{2}$-inch **T** *d* at the upper end of this pipe is connected at one end of the run with a $1\frac{1}{4}$-inch check-valve *e* opening inwards, and at the other end with a $\frac{3}{4}$-inch check-valve *f* that opens outwards. The valve *f* is connected with the air chamber through the pipe *g*.

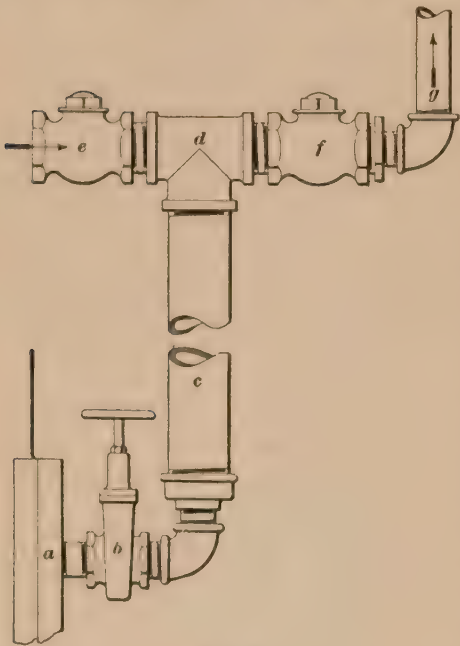


FIG. 19.

This air pump is operated as follows: When the pump is working, open the valve *b* to fill the pipe *c* with water; then partially close *b* until the check-valves *e* and *f* begin to work. This is easily determined by the click of the valves when seating. Its working may be described thus: When the valve *b* is opened, water fills the pipe *c* from the pump cylinder *a* during the discharge stroke of the pump. By partly closing *b* when *c* is full, the pump during the suction stroke will draw a part of the water from *c*, and air will flow in through *e* to take its place. During the next discharge stroke of the pump, more water is forced into *c*, driving the air out through *f* and *g* into the air chamber. If *b* is opened too wide, all the water will be drawn out of *c* during the suction stroke and air will be drawn into the pump cylinder from *e*; but by properly regulating the opening, a column of water is kept in *c*, which acts as a piston that moves with the strokes of the pump and pumps air into the air chamber.

28. Alleviator.—When pumps work under pressures greater than that due to a 350-foot lift, air chambers are not of very much service, owing to the fact that the air escapes from the air chambers either through the pores of the iron or at the joints, or it is absorbed and carried off by the water; in such a condition an air chamber gives the pump no relief whatever. To obviate this defect **alleviators** are used. An alleviator is shown in Fig. 20. It consists of a plunger *a* working through a water-packed stuffingbox. On top of the plunger are arranged springs that may be in the form of rubber buffers or helical coil springs. In the type shown rubber buffers *b*, *b* are used, which are confined by the tie-rods *c*, *c*, the yoke *d*, and the plates *e*, *e*. When the pressure in the pipe exceeds the working pressure, the plunger *a* is forced out through the stuffingbox and relieves the pump of the shocks that would otherwise occur. Alleviators may be placed anywhere on the delivery pipe, but are preferably placed in such a position that the direction of the moving water is in line with the plunger *a*.

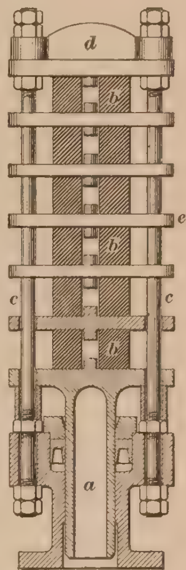


FIG. 20.

SUCTION AIR CHAMBERS.

29. Purpose.—With a long suction pipe or a pipe with numerous bends and valves, the resistance to the flow of the water through it will be considerable, and a great deal of force will be required to start and stop the water in it with each stroke of the pump. In some cases the force required is so great that the pressure of the atmosphere is not sufficient to set the column of water in motion quickly enough to fill the pump chamber as fast as the piston moves. This makes the action of the pump imperfect and causes a

severe blow, called the **water hammer,** when the piston again meets the inflowing water.

30. The difficulty mentioned in Art. **29** can best be remedied by the use of a chamber, called a **vacuum chamber** or a **suction air chamber,** attached to the suction pipe as near the pump as possible. In its general form a vacuum chamber resembles an air chamber, but the pressure in it instead of being greater is always less than the atmospheric pressure. When the pump is drawing water, the air in the vacuum chamber expands and forces the water below it into the pump; at the same time the pressure of the atmosphere forces water in through the suction pipe to balance the reduced pressure in the vacuum chamber. The vacuum chamber is again partly filled and the air in it is compressed during the discharge stroke of the pump. It thus acts as a reservoir that receives from the suction pipe a nearly steady supply, which is given up intermittently to the pump.

31. Special Form of Suction Air Chamber.—Fig. 21 shows a special form of a suction air chamber in diagram-

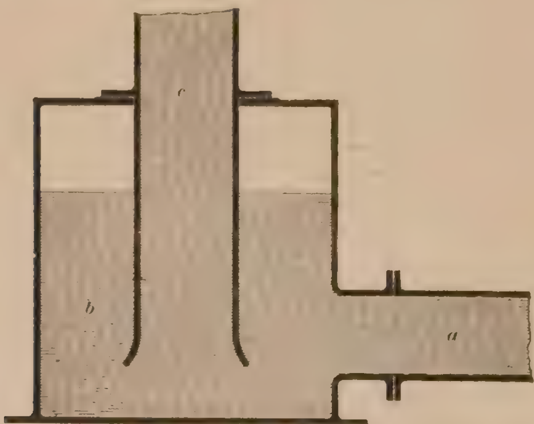


FIG. 21.

matic form. The suction pipe *a* connects to a suitable chamber *b*, which has a tube *c* projecting downwards to

within a short distance of the bottom. The tube *c*, which is called a **draft tube,** connects to the pump chamber. When water first flows into the chamber *b*, it entraps some of the air as soon as the water seals the bottom of the draft tube; this air is then compressed while the water flows up the draft tube, and by its expansion and compression permits a steady flow in the suction pipe.

32. Size of Vacuum Chambers.—For ordinary cases, the vacuum chamber may be made half the size of an air chamber working under the same conditions. A good rule is to make the cubic capacity of the vacuum chamber for a single pump twice that of the displacement of the piston for a single stroke.

33. Location. — Suction and delivery air chambers should, if possible, be placed at a bend in the pipe and close to the pump and in such a position as to be in line with the flow of water in the pipe. If placed at right angles to the flow of water, as in Fig. 18, their efficiency is somewhat impaired. Both suction and delivery air chambers should be provided with glass water gauges so that the height of the water can be determined at a glance. It is not customary to provide the air chambers of small pumps with water gauges.

PUMP FOUNDATIONS.

GENERAL CONSIDERATIONS.

34. The foundation for pumping machinery depends entirely on the type of pump. Generally speaking, much less foundation is required than for steam engines occupying about the same space. Direct-acting duplex pumps probably require the least foundation of any kind of steam pump, for here the piston and plunger motion is almost opposite and the balancing of the machine in line with the plunger motion is complete, and the strains due to reversing are contained almost wholly within the machine itself. Small duplex-pump foundations are made of a solid mass of brick or

concrete, while large pumps are often set on separate piers, one for the water ends and one for each pair of steam ends in case of a duplex, compound, or triple-expansion engine. Of course, the foundations must go down to sufficiently hard soil to bear up the weight of the pump, or if the soil be loose sand or gravel, the foundation must be spread out sufficiently to insure the pressure not exceeding, say, 1 ton per square foot. The foundation should go deep enough to allow the surrounding soil sufficient hold upon it to keep it firm and steady. The minimum depth for a small pump should not be less than 2 feet. Single-cylinder pumps require a somewhat heavier foundation than duplex pumps, owing to the greater shocks to which they are subjected.

35. Crank-and-flywheel pumps require considerably more foundation than direct-acting machines, on account of the much higher speeds possible and the weight and lack of balance of the reciprocating parts. Crank-and-flywheel pumps of the controlled-valve type, as the Riedler pumps, which usually run at a high speed, require foundations fully as heavy as those for steam engines of equal size.

MATERIAL AND FOUNDATION BOLTS.

36. Foundations should be built of hard brick laid in cement mortar, concrete, or, in the case of large pumps, of stone, if it can be readily secured. All pumps should be held down by foundation bolts. In the case of small pumps the bolts are provided with a steel or wrought-iron plate washer built solidly into the foundation, while large pumps have tunnels or pockets for access to the lower foundation washer and nut. If the foundation bolts are built in solid, box washers should be used.

FOUNDATIONS FOR LARGE PUMPS.

37. In the case of large vertical pumping engines, the masonry required to form the pump pit and to support the superstructure is of ample mass for all foundation purposes; in fact, large arched chambers and tunnels are often used to

save foundation materials in this class of pumping engine. These large pumping engines are often located at or near a water supply where the soil has not sufficient rigidity to support the weight. In this case piling must be resorted to, on which the foundation proper is constructed.

USE OF FOUNDATION TEMPLET.

38. A foundation templet should always be used in which the foundation-bolt holes are carefully laid off, preferably from the actual castings, and the various heights of bosses or thicknesses of casting through which the bolts pass are marked. The templet should be carefully set with reference to the suction and delivery connections, so that when the pump is set up, the fittings and pipes will connect up properly. In large pumps it is customary to arrange the pipe connections in such a way that a short space is left between the piping and the pump. This space is then measured after the pump and piping are in place, and a distance piece is made to suit the measurement and then put in place.

FOUNDATIONS FOR SMALL PUMPS.

39. Small pumps of the single-cylinder and duplex type are usually provided with two points of support only, one of which is rigidly bolted to the foundation, while the other is left free. This prevents the pump being thrown out of line, if properly constructed originally. When both the steam and water ends are bolted down, care must be taken not to twist or throw the pump out of line. In making the steam and water connections, the pipes should come fair to their connections and should not be sprung into place. Stresses on the pump structure due to winding foundation surfaces and sprung pipe connections should be guarded against, particularly with steam-thrown valves, as these are very sensitive and must be perfectly free. Any slight springing of the valve chamber will bind the valve and prevent its operating.

PIPING.

SUCTION PIPING.

40. Location of Pump in Respect to Supply.—Before a pump can be properly located, the location of the source of supply of the liquid to be pumped must be taken into consideration. Since the atmospheric pressure of 14.7 pounds to the square inch will balance a column of water 34 feet high, it is evident that with that atmospheric pressure the pump must not be placed more than 34 feet vertically above the surface of the water to be pumped. But since a perfect vacuum cannot be obtained by mechanical means, and since the flow of the water is retarded by friction in the pipes and passages, the limit of vertical lift by atmospheric pressure is reduced to about 28 feet at sea level in actual practice. The actual lift, precisely as the theoretical lift, varies with the atmospheric pressure, and hence will become smaller with an increase of altitude above sea level, since the air becomes lighter and its pressure less.

41. Run of Suction Pipe.—The pump should be placed as near the source of water to be pumped as is possible, both vertically and horizontally. The suction pipe should be as straight as possible; if bends are necessary, they should be made by bending the pipe to a long radius or by using long-turn fittings. The suction pipe should be one diameter from end to end; all enlargements or reductions in size tend to disturb the uniform flow of the water so essential to a proper filling of the pump chamber. If from necessity the suction pipe is very long, it will be well to increase the size somewhat; the reduction at the pump chamber should then be made by a long conical fitting. For ordinary service pumps the diameter of the suction pipe should be such that the velocity does not exceed 200 feet per minute, assuming that the flow of water is constant. If the vertical lift be high, a suction air chamber should be provided; this will

add much to the uniformity of the pump supply. A foot-valve should also be provided when the lift is high.

42. Foot-Valves.—A foot-valve is a check-valve placed at the lower end of the suction pipe below the water level in the source of supply and opening towards the pump. Its purpose is to prevent the suction pipe emptying while the pump is at rest and to prevent the water in the suction pipe slipping back while running. When the water flows to the pump by gravity, a foot-valve is superflüous; but when the water is lifted by suction it is often fitted, since it will insure a prompt starting of the pump, providing that it is tight enough to hold the water in the suction pipe. In very cold weather and in exposed locations, the foot-valve constitutes an element of danger when the pump is out of use, since it prevents the emptying of the suction pipe. The water in the latter may freeze and burst the pipe. To prevent this, a drain may advantageously be fitted to the lower end of the suction pipe, which is used in cold weather to empty the pipe if the pump is to stand idle for a long time.

43. When foot-valves are used, a relief valve may advantageously be placed on the suction pipe. Generally, the suction pipe is made considerably lighter than other parts of the pump, and if the suction valves should leak when the pump is standing or if the priming pipe be left open, the full pressure of the delivery water will come on the suction pipe and foot-valve, which are not usually designed to withstand such pressures. The relief valve, which should be set to relieve the pipe at a pressure well within its safe strength, prevents overstraining of the suction pipe from this cause. Foot-valves should be chosen with the greatest care; they should be simple and, preferably, of the weighted-lift type or clack valve, and should have at least 50 per cent. excess of area over the suction pipe.

44. Settling Chamber.—If the water to be pumped is gritty or contains foreign substances, a settling chamber is sometimes used, especially when pumping water holding but

a small quantity of sand in suspension. This consists of an iron box conveniently arranged in a horizontal pipe. It is usually of large relative capacity, a settling chamber for a 2-inch pipe being 2 feet × 2 feet × 3 feet long. The pipes enter and leave from opposite sides and near the top. The increased volume of the large box allows the water to move very slowly across the box, giving the suspended sand time to settle to the bottom. The settling chamber should have a removable cover for the purpose of removing the settlings. This device is used on small pumps working on artesian wells.

45. Suction Basket and Strainer.—More universal arrangements for keeping back foreign matter from the working barrel of the pump are the **suction basket** and the **strainer.** The suction basket is usually placed on the bottom of the suction pipe and consists of a box variously shaped and perforated with strainer holes or provided with screens. The suction basket so placed is being replaced by a different form of strainer, which consists of a chamber placed in the suction pipe, located in an accessible position and provided with strainer plates so made that they can be readily removed for cleaning. This strainer is sometimes connected directly to the pump, but it should not be so placed that it will interfere with removing the water-cylinder heads. A short piece of pipe between the strainer and pump nozzle will avoid this interference. The objection to the suction basket on the bottom of the suction pipe is its inaccessibility for cleaning and inspection, a feature that is overcome by the strainer.

DELIVERY PIPING.

46. Run and Valves.—While the suction pipe is very important and must be most carefully laid out and has much to do with the location of the pump, the delivery pipe should not be neglected. A careful adjustment between the supply and delivery pipes should be made in order to produce the

best effect of the whole plant. The delivery pipe should as far as possible be a plain, straight pipe from pump to terminal; when bends are necessary, they should be by as long sweeps as possible. A gate valve or check-valve should be placed near the pump. The check-valve serves the double purpose of relieving the pump of pressure when starting up, allowing it to take hold of the water more quickly, and also holds the water back from the pump when inspection and repairs to the water end are necessary. If a check-valve is not used, a gate valve should be placed at or near the pump delivery to hold back the water in case of repairs to the pump end or accident. This valve should always be a straightway gate valve giving the full clear opening of the pipe.

47. Velocity of Flow.—The velocity of the water flowing through the delivery pipe for direct-acting pumps should not much exceed 330 feet per minute, while for large crank-and-flywheel pumping engines the velocity of water in both suction and delivery pipes is about 300 feet per minute. If the suction pipe is made small, the pump will fail to fill and the plunger will strike the incoming water on its return stroke, producing a violent and dangerous shock. If the delivery pipe is made small, the cost of power required to force the water through the pipes at a high velocity will very quickly overrun the interest and depreciation on a larger pipe.

AUXILIARY PIPING.

BY-PASSES.

48. Water-End By-Pass.—**By-pass pipes** are pipe connections from above to below the delivery-valve deck and are of much more use on crank-and-flywheel pumps than on direct-acting machines. In the case of compound pumps, when starting up, the force of the full steam pressure on the high-pressure piston is not sufficient to move the plungers

against the resistance due to the head of the water in the delivery pipe; but by opening the valve (which, by the way, should always be a gate valve) in the by-pass piping, the pressure on the plungers is relieved for a sufficient number of strokes to allow the steam to reach the low-pressure piston, when the combined force of the two pistons will do the work and the by-pass pipe can be closed.

49. By-pass water pipes have another function on crank-and-flywheel pumps. Unless these machines are fitted with very large flywheels, their limit to slow running is often not as low as desired. By opening the valve in the by-pass pipe, part of the water can be returned to the pump chamber and the amount of water actually pumped reduced to any desired quantity permitted by the size of the by-pass. It should not be overlooked that this is accomplished at a very considerable loss of efficiency, because it takes the same power to move the by-pass water as it does to do the actual pumping, comparing equal quantities. By-pass pipes are usually made $2\frac{1}{2}$ per cent. of the plunger area.

50. Steam-End By-Pass.—It is common practice to fit the steam cylinders with by-pass pipes, allowing high-pressure steam to act on the low-pressure piston in starting, but these pipes are usually so small, compared with the diameter of the low-pressure piston, that the by-pass is unable to hold any pressure behind the low-pressure piston when it is moving. By-pass steam pipes have their proper use in warming up the low-pressure cylinder and connections, and in the case of crank-and-flywheel pumps to move the high-pressure crank off the dead center.

PRIMING PIPE.

51. The **priming** or **charging pipe** is a small pipe run from the delivery pipe beyond the check-valve or delivery gate valve to the suction chamber of the pump. It is particularly useful in the case of long suction lifts to fill the working chamber and suction pipe with water, taking up all

clearances and helping the pump to take hold of the water quickly. This pipe may be from $\frac{3}{4}$ of 1 per cent. to 1 per cent. of the area of the plunger; its size is a matter of little importance, but it should be large enough to fill the suction pipe and pump chamber in a reasonable time, which will depend somewhat on the size and design of the pump chamber and the length of suction pipe. A pipe much larger than 1 per cent. of the plunger area will be required in the case of long inclined or horizontal suction pipes.

WASTE DELIVERY PIPE.

52. A **waste delivery** or **starting pipe** that can be led into any convenient place of overflow should be provided so that the pump, in starting, can free itself of air, for it often happens that a pump refuses to lift while the full pressure against which it is expected to work is resting on the delivery valves, for the reason that the air within the pump chamber is not dislodged but only compressed and expanded again by the motion of the plunger. A pump in this condition is said to be **air bound.** It is well in this case to run with the delivery pipe empty until the air is expelled and the water flows into the suction end of the pump. The waste delivery pipe is fitted with a valve and connected to the delivery pipe close to the pump. When the water flows to the pump and is discharged into the delivery pipe, the valve in the waste delivery pipe is to be closed.

AIR DISCHARGE VALVES.

53. When a check-valve is not used in the delivery pipe and the space between the suction and delivery valves is large and the delivery pipe is full of water, the pump will often refuse to start the water in the suction end, owing to compressed air being trapped between the water in the delivery deck and suction valves. Air discharge valves, each composed of a globe valve and a check-valve, may then

be used on each head, the check-valve opening to the atmosphere, thus permitting the escape of air but preventing its entrance when the globe valve is open. The globe valve is closed when the pump is working properly, as shown by water coming from the check-valve.

GENERAL PIPING ARRANGEMENT.

54. Fig. 22 shows a good arrangement of a pump in relation to the water supply and of the pipe connections. The

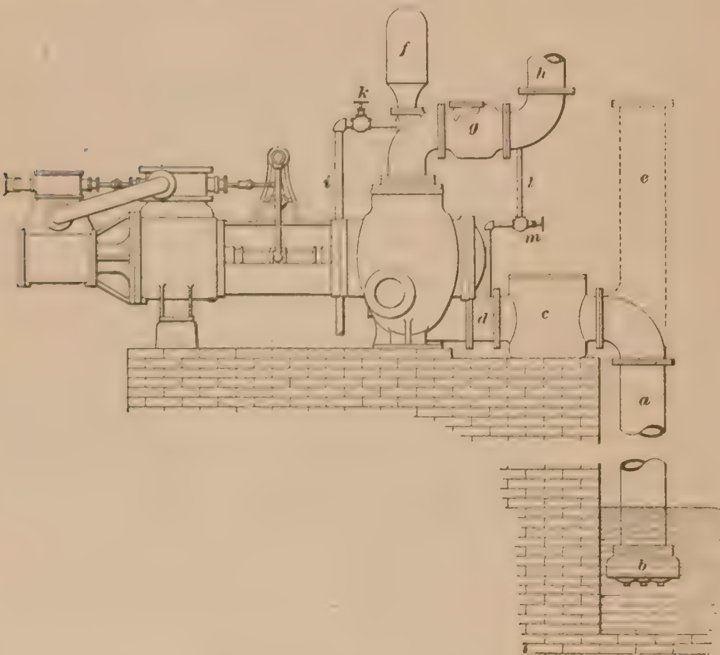


FIG. 22.

suction pipe *a* is fitted with the foot-valve *b* and has a strainer *c* placed close to the pump, from which it is separated by the short distance piece *d*. When the vertical lift is short, say not over 10 feet, and the pump is placed close to the source of supply, a suction air chamber is seldom necessary, but when the lift exceeds 10 feet or when the pump is

at some distance from the water supply, a suction air chamber becomes a necessity. With a vertical suction pipe as shown, the suction air chamber may be made as shown by the dotted lines at *e*. An air chamber *f* is placed on the delivery between the delivery check-valve *g* and the delivery valves. The waste delivery or starting pipe *i* is connected to the delivery between the delivery valves and the delivery check-valve *g*. It is fitted with the valve *k*. The delivery pipe *h* is connected to the suction pipe close to the pump, in this case to the distance piece *d*, by the priming pipe *l*, which is fifted with the stop-valve *m*.

PROVISION FOR DRAINAGE.

55. Proper drain pipes and drain valves should be provided for all parts of the pump, the pipe connections, strainers, etc., in short, for all parts in which water may remain when the pump is not in use and will give trouble by freezing.

Provision for draining the suction valve deck and delivery valve deck is sometimes made by drilling a small hole through the decks; this practice, while simple and cheap, leads to a loss in efficiency, however, since some of the water is constantly flowing back into the suction chamber.

PUMP MANAGEMENT.

INTRODUCTION.

56. If a pump has been properly selected for the service and has been properly designed, built, and erected, it should perform its work without any trouble. All pumps when new are stiff and cranky in their actions, particularly direct-acting pumps. They should be run slowly for a considerable time, and many defects in their action which at first gives rise to alarm will then gradually disappear. Crank-and-flywheel pumps act more smoothly from the start, but do

not come to a proper bearing more quickly or quite as quickly as the direct-acting pump. Crank-and-flywheel pumps usually require considerable skill and study to reduce them to successful working order, as conditions arise that further disturb the lack of harmony between the flywheel and water, and it often taxes the skill of the experienced engineer to make an amicable adjustment between the two opposing forces.

57. Having reduced the pump to satisfactory operation, the attention of the operator should be directed to its maintenance at the least possible expenditure. Each item of expenditure should be separated from the whole and studied independently for the purpose of reducing it to a minimum consistent with the proper maintenance of the plant. The expenditure should at all times be regarded as the item by which interest or dividends are being earned and should not be allowed to become greater.

58. Losses in efficiency arise from wear, from loss of proper adjustments, and from the wrong timing of the various movements that control the distribution of steam, by leakages, by decreased mechanical efficiency due to lack of alinement, by accumulations of foreign matter on and in condenser tubes, suction strainer, and foot-valves, suction and delivery pipes, and in many minor directions. In many plants it is of the utmost importance that they should not be interrupted; it is then the duty of the engineer to predict all possible events that might cause an interruption and have a well-planned line of action prepared so that he may act quickly and with decision to the end of keeping his plant always at work and at the highest efficiency. This plan of action will entail considerable work, study, and, perhaps, some expense in preparation to meet possible contingencies that may never happen; nevertheless, it is well to be ready for any emergency when handling steam machinery and particularly steam pumping engines.

59. In the management of pumps it must be considered that nearly every installation has its peculiarities, some of

which are sometimes not discovered until after the machine is put in service and then perhaps require expensive additions and alterations to meet them. An exhaustive study of existing conditions and resultant conditions when the pump starts to work cannot be too strongly urged.

STARTING PUMPS.

IMPOSSIBILITY OF SPECIFIC RULES.

60. Pumps differ so much in their construction and design that it is entirely impossible to lay down specific rules that will be applicable to every pump. For this reason only *general* rules are here given, which must be modified by the pump attendant to suit every specific case.

GETTING A PUMP READY.

61. Getting Up Steam.—Considering a new steam pump, after it has been properly erected on a suitable foundation and all the pipe connections have been made, the first step in starting the pump is to get up steam in the boiler or boilers in the same manner as is done with boilers supplying steam for any other purpose.

62. Since the boilers are generally in charge of the same person that attends the pump, the general treatment of the pump and the boilers, while steam is being raised, will be considered together. After the steam piping is in place, but before it is finally connected to the pump, all valves in it should be opened wide; while steam is being raised the pistons and valves should be removed from the steam end of the pump so that there is a clear passage for the steam from the boiler to the exhaust after the steam pipe has been connected to the pump.

63. Blowing Out the Steam Piping.—The fires should be started very slowly under the boiler; all the binding

bolts throughout the boiler setting should be perfectly loose and free. If this precaution is neglected, buckstaves or cast-iron fronts will be broken by the expansion of the setting. The guy rods on iron stacks should also be slacked off; in fact, every part that will expand when the plant is started up should be liberated. Before the steaming stage is reached, large volumes of heated air will be driven through the pipes, warming them up gradually. When steam begins to rise, it should be allowed to blow through the piping and valves quite liberally, the object being to clear the piping of sand, grit, and all other foreign matter collected therein during erection. The piping having been blown out thoroughly, steam is shut off and the piping is then connected to the pump.

64. Blowing Out the Cylinder.—When the pressure in the boiler has been raised to the working pressure, the cylinder heads should be put on, still leaving the pistons and valves out of the cylinders. The stuffingboxes should be closed, which is most conveniently done by placing a piece of board between the stuffingbox and the reversed gland and then setting up the nut on the stuffingbox studs. When the gland is drawn home by a nut outside of it, a circular piece of pine board may be placed between the end of the gland and the inside of the nut in order to close the opening through which the piston rod passes. The steam may now be turned on the main steam pipe leading to the pump; by opening the throttle valve wide at short intervals, the sand and scale in the ports and other passages and spaces of the steam end can be blown out. After the cylinders have been blown out, the heads and covers should be removed, and all foreign matter blown into the corners and chambers of the cylinders should be removed by hand. The pistons, valves, cylinder heads, and other covers can now be put in place.

65. The blowing out of the pipes and cylinders after erection is often neglected or but imperfectly done, with

serious consequences to the machine; it cannot be too thoroughly done, particularly in that type of pump where the steam ports and exhaust ports are on top, for in this particular case the sand and grit are deposited in the bottom of the cylinder for the piston to ride upon. If more attention were paid to the thorough cleaning of all steam spaces, we would hear less of cylinders and pistons being cut.

66. Keying Up.—If the pump is of the crank-and-flywheel type, it should be turned a complete revolution by hand to insure that everything clears properly and that no tools or materials used during construction or erection have been left within the machine. The adjustment of all journals, pins, and bearings should then be made. With gib and key ends, it is usual to drive down the key with a soft hammer (lead hammer) until it is home, mark it, drive it back, and then tap it down to within $\frac{1}{8}$ inch of the mark. With wedge ends the wedges usually have an inclination of $1\frac{1}{2}$ inches per foot and the adjusting screw 8 threads per inch. The wedge is drawn up solid and then the adjusting screw is turned back about 20° and locked. Bolted connecting-rod ends are allowed about $\frac{1}{200}$ inch play, using liners and setting the bolts up solid. Main bearings can be adjusted best when the machine is in motion.

67. Packing Rods and Stems. — The packing of all rods and stems is the next step. If fibrous packing is used, the boxes should be filled full and the glands tightened down very moderately. The tightening of the glands can best be done when steam is on and the machine is in motion, when they should be tightened only sufficient to stop leakage and no more. When excessive tightening is required to stop leakage, the packing should be completely renewed. Some pumps are fitted with metallic packings. These packings are usually fitted up by specialists who fully guarantee them, and their directions for use should be carefully followed; in case of failure or unsatisfactory results, the makers should be consulted.

68. Oiling.—The oiling of the machinery is the next step and is a very important one. All rubbing surfaces should be provided with suitable oiling devices appropriate to the particular place and service. The quality of oil should be carefully selected to suit the velocity and pressure of the rubbing surfaces on which it is used. For use within the steam cylinder, heavy mineral oil is the only oil capable of withstanding the high temperature, and in starting up new pumps only, the best quality should be used, regardless of price. A liberal use of this oil for the first month will go far towards reducing subsequent oil bills.

69. The pumping engine, unlike many other types of engines, must often run continuously and without interruption for a month or even longer at a run. This requires that all oiling devices be so arranged that they can be supplied and adjusted while the machine is in motion. It is a good plan to provide two separate sets of oiling systems for all the principal journals, the idea being that if one fails the other can be used while the disabled one is being overhauled. All oil holes should have been filled with wooden plugs, bits of waste twisted in the hole, or some other protection, while the machine was being erected. These should now all be removed and all oil holes and oil channel thoroughly cleaned out. Bearings should be flooded with oil at first to wash out any dust or grit that may have reached the rubbing surfaces.

70. Having turned the machine by hand and inspected all locknuts, setscrews, and clamp screws, the engine may be put under steam. If provided with hand starting gear, this should be used for a sufficient number of turns to make sure that the machine is free from water that may have accumulated in the pipes or clearance spaces. All drain cocks should be wide open when starting and relief valves should be adjusted to blow at the proper pressure. If the engine is condensing, connections from the exhaust port to the condenser should be made absolutely tight. If an independent condenser is used, it should be started before

the main pump is started and a vacuum obtained in advance.

71. So far only the steam end of a large crank-and-flywheel pump has been considered. With the direct-acting single or duplex steam pump, the same general method of procedure should be followed. It may be mentioned here, incidentally, that the direct-acting pump is not so liable to an accident in starting as the crank-and-flywheel pump on account of the absence of kinetic energy stored up in a moving flywheel. This energy when given out by reason of an obstruction in the water end that prevents the free passage of water will greatly increase the pressure, especially when the obstruction occurs near the dead-center positions of the crank. The increased pressure thus produced may easily run up high enough to burst the water end.

72. Using the Dash Relief Valves.—In starting a direct-acting pump when dash relief valves are fitted, they should be closed in order to keep the pistons as far from the heads as possible, for in new installations the unexpected is likely to happen at the water end, and to prevent danger of a breakdown, as in case of a sudden lunge of the pistons, all the margin possible to keep them from striking the heads should be gained.

73. Condition of Water End When Starting.—Assuming that the plungers and plunger rods are packed and the plunger grease cups filled, the water end should be ready to start; if the machine is compound or triple-expansion, the water by-pass valves must be opened until the machine has made a sufficient number of strokes to bring the intermediate and low-pressure cylinders into action, when the by-pass valves should be closed. The suction pipe from the foot-valves to the delivery valve deck must be absolutely tight; anything short of this will cause the water end to refuse to work satisfactorily. All the suction valves and delivery valves should seat fairly and tightly. Care must be taken that there is no obstruction in the delivery pipe, such as a

closed valve, as pumps usually have sufficient margin in the driving force over the resistance to burst the water end, particularly if the momentum of a flywheel be added to it.

74. Pressure gauges should always be attached to the suction and delivery pipes, and they should be carefully watched during the process of starting, as trouble at the water end will be promptly recorded by the gauges. The lower end of the suction pipe should be kept well under water, as a slug of air taken into the pump may cause a violent jumping and in a direct-acting pump possibly a striking of the steam pistons against the heads.

75. Watching the Air Chamber.—The delivery air chamber should be carefully watched during the starting and running. This should be provided with a gauge glass showing the height of the water and extent of the pulsation. The air chamber should be charged with air when the air in the chamber is lost, as shown by the rise of the water in the gauge glass. Large pumps are usually supplied with an air charging pump that is attached to and driven by the main pump, or an arrangement of pipes and valves is sometimes improvised for this purpose. In very large pumping plants, an independent air compressor or locomotive air pump is often used for this service. A very good idea of the internal working of the pump can be obtained by placing the ear against the pump chambers; the seating of the valves can then be distinctly heard, and if there is any leak past either the suction or the delivery valves, it, too, is quite audible

DEFECTS IN PUMPS.

SUCTION-END TROUBLES.

76. The most common causes of pump failures are leaks below the suction valves. These may be at the joints or along the suction pipe or in the pump chamber, and may be due to imperfect connections, leaky chaplets, shifted cores, blowholes, corrosion, or cracks from frost.

77. Small leaks in the suction end which are not sufficient to cause entire failure will cause the piston to jump, i. e., move suddenly, during the first part of the stroke. Leaky valves and plungers reduce the capacity of the pump; if this is the case, they should immediately be refitted and repacked. It is always best to have hot water flow to the pump by gravity; if it is necessary to lift it and the pump works with a jerky action, the lift is too high for the temperature, and one or the other must be reduced. In pumping from wells, care should be taken that the pump is near enough to the water to prevent the water falling below the maximum lift by suction.

78. If the pump pounds soon after the beginning of a stroke, when running fast, it shows that the pump chamber is not filling and that the plunger is striking the incoming water on its return stroke. A suction air chamber will help to remedy the evil. Obstructions under the suction or delivery valves will cause a very decreased output or total failure. A suction strainer or end of suction pipe becoming embedded in sand or clogged with foreign matter will cut off the supply from a pump.

79. Air pockets under the delivery valve deck, caused either by bad design or a shifting core, will very much reduce the capacity and efficiency of a pump. The effect of the air pocket is to entrap air, which is compressed to delivery-water pressure and expands again on the suction stroke. If the relative capacity of the pocket to the plunger displacement is sufficient, the entrapped air will expand to atmospheric pressure, reducing the suction lift to zero; this defect, however small, will always reduce the suction lift and is not easy to remedy; its existence should always be cause for the rejection of a pump.

80. Pounding in pumps is sometimes caused by the water lagging behind the plunger, due to the friction of a small, long, horizontal suction pipe. When suction pipes have a long horizontal run, they should be one or two sizes larger.

DELIVERY-END TROUBLES.

81. Pumps sometimes fail when the full head is resting upon the delivery valves by the air between the suction and delivery valves being expanded and compressed by the motion of the plunger. Air cocks should be provided close up under the delivery decks for discharging the air until the plungers have caught the water. If only a simple cock is fitted, it must be opened during the delivery stroke only and closed shortly before the suction stroke commences. This is to be repeated until a steady stream of water issues from it during the delivery stroke. An automatic air valve, which is simply a small spring loaded valve opening outwardly and closing automatically during the suction stroke, is preferable; this valve should be secured to its seat after a steady stream of water issues during the delivery stroke. Violent jarring and trembling of the pump arises from the delivery air chamber becoming filled with water. It should be recharged with air by means of the air-charging pump, a near-by air compressor, or by a hand air pump.

STEAM-END TROUBLES.

82. The steam end of pumps should not be taken apart needlessly, especially the steam end of direct-acting pumps with steam-thrown valves, as their action is quite complicated, and a very slight misadjustment will cause a failure. If at any time it becomes necessary to dismantle the pump, all the parts, if not already marked, should be plainly marked with steel letters or numbers, rather than with a prick punch or chisel, and suitable gauges, by which all parts can be returned to their correct relative positions, should be made, if this is deemed advisable. In many duplex pumps there are very slight differences in the two sides; for instance, the crossheads that drive the valve levers are not keyed in exactly the same position on the piston rods and the rods are not interchangeable; the pump will not run successfully if they are interchanged. In some pumps with steam-thrown valves, the valve chests are bolted to the cylinders,

and are reversible so far as fitting and bolting goes, but the auxiliary ports are not reversible and will be shut off in both valve chest and cylinder by reversing the chest. In placing the gasket between the valve chest and cylinder of pumps with steam-thrown valves, care should be taken to cut passages through the gasket for the auxiliary ports. The valve levers, pins, and all connections between the piston rod of one side of a duplex pump and the valve of the opposite side should be kept in good condition, as the failure of these parts will cause a serious accident.

83. On duplex pumps the amount of lost motion between the valve stem and the valve should be very carefully adjusted; too little lost motion will cause short stroking, while too much will allow the pistons to strike the heads. If the pistons strike the cylinder heads, the dash relief valves, if fitted, should be closed until the stroke is shortened sufficiently for the pistons to clear the heads. If the stroke becomes too short, the opposite course should be followed If no dash relief valves are fitted, the lost motion should be made smaller in case the pistons strike the heads.

84. When a compound pump is fitted with a cross exhaust and it is seen that the pump is unable to complete its full stroke, the valve in the cross exhaust should be opened.

TESTING PUMPS FOR LEAKAGE.

85. Testing the Suction Pipe.—Leaks are the most troublesome and most frequent sources of loss of efficiency in pumping machinery. Leaks in the suction pipe or suction system affect the pump most and often cause its complete failure. These leaks can sometimes be detected by the ear, or the flame from a common tallow candle will often locate a leak in the suction by being drawn towards the hole by the air. Sometimes these leaks are very numerous, but so small that any one of them would be difficult to locate and be of small importance; at the same time, their combined effect may be sufficient to seriously affect the working of the

pump. The best way to locate these leaks, which may be at the joints or along the body of the pipe, is to stop up the inlet end of the pipe, uncover it completely, and then put a water pressure on it, say from 40 to 50 pounds per square inch. Any leaks, however small, will then be readily detected. The suction pipe should always be tested for leaks before it is covered, if laid in a trench or otherwise made inaccessible, because it must be made tight before the pump will work successfully.

86. Delivery Pipe Leaks.—Leaks in the delivery pipe, while common and at times more difficult to remedy than leaks in the suction, are plainly evident. They do not affect the action of the pump or its efficiency to any extent, the loss being exactly proportional to the magnitude of the leak.

87. Repairing Leaky Pipes.—Probably the most satisfactory method of procedure in case a leaky section of pipe is discovered is to discard it and replace it with a new one. Circumstances, however, do not always permit this to be done, and then temporary repairs should be made. The manner of making the repair obviously depends on the position and extent of the leak and calls for the exercise of judgment and some skill.

88. Small leaks in the form of pinholes in the suction pipe can generally be stopped effectually by a thick coat of red-lead putty spread over the pipe where the leaks occur. This should be covered with several layers of canvas covered on both sides with red-lead putty and wound as tightly as possible around the pipe. The canvas should then be secured by wrapping it with strong twine or annealed copper wire, put on as tightly as possible. If the suction pipe is split, it is usually well to cover the split part with a piece of sheet metal, preferably sheet lead, bent to the curvature of the pipe and put on with red-lead putty. The canvas should be wrapped over this.

A permanent repair in case of pinholes can be made by drilling out the pinhole with a twist drill and tapping out

the hole. A closely fitting threaded plug of soft steel or wrought iron is then screwed in and the end riveted over.

89. Small pinholes in delivery pipes can often be stopped up by the same means given in Art. **88** for suction pipes. If the leak is extensive, however, it will generally be necessary to use a **pipe clamp.** Such clamps may be made in a good many different ways, according to the location and extent of the leak and the facilities for repair. One of the simplest pipe clamps is shown in Fig. 23. It consists simply of a piece of sheet iron or sheet steel of sufficient width to cover the leak and bent to the form shown. A piece of sheet packing, which may be covered with red-lead putty to advantage, is placed over the leak and the pipe clamp is then placed over this and the ends drawn together by the bolt shown.

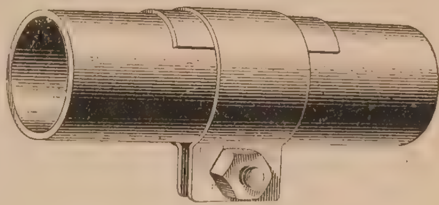


FIG. 23.

The clamp shown in Fig. 23 is only adapted for small pipes. For large pipes the clamp must be made in two halves.

90. Testing Air Chambers.—Air chambers must be absolutely tight. They are usually tested by closing all openings and then pumping air into them until the working pressure is reached, as shown by a pressure gauge. After 24 hours this gauge should show no reduction of pressure. If the air chamber does not pass this test, the leaks may be discovered by filling it with water subjected to the working pressure. If there are a number of leaks, the chamber should be condemned; if only a few small leaks exist, they can usually be effectually stopped by drilling a hole at the leak and screwing in a plug.

91. Leakage of Pistons and Plungers.—The plungers of inside-packed or center-packed plunger pumps should be

tight themselves, besides making a tight joint through the stuffingboxes, in order that water may not pass from one side to the other. The manner of testing will depend on their design, the general method of procedure being the subjecting of one side of the plunger to an air pressure or hydrostatic pressure at least equal to the working pressure. If leaks are discovered, judgment has to be used as to the manner of repairing them or whether to condemn the plunger. In some designs of inside-packed and center-packed pumps with closed hollow plungers, the weight of the plunger is so proportioned to its displacement as to relieve the stuffingboxes of nearly or quite all of its weight; it is then important that they be absolutely water-tight.

92. Leakage Past Pistons and Plungers.—With piston pumps and inside-packed plunger pumps there is liable to be considerable unnoticed leakage. If it is extensive, it can be heard by placing the ear against the pump chamber. It is best with this style of pump to make regular inspections for leakage past the plunger or piston, providing suitable pipes and apparatus by means of which pressure can be put on one side of the packing or piston while the other side is exposed for inspection. With outside-packed plungers there can be no unobserved leaks past the plungers, and this is the principal reason for their use.

93. Leaks Past the Valves.—Leaks past the suction and delivery valves can readily be tested when the piston or plunger is being tested for leaks past them. The delivery and suction valves should be tested separately; the fact that the column of water in the delivery pipe does not drain out while standing is not proof that both sets of valves are tight, since either set will support the water while the other set may be leaking badly.

94. To test the suction valves for leakage, disconnect the suction pipe or take any other convenient steps that will allow the leakage to be seen. Fill the delivery pipe full of water, having removed enough delivery valves to allow the pressure to reach all the suction valves, and observe which

valves, if any, are leaking. When there is a valve in the delivery pipe, this may be shut and water pumped into the pump cylinder with a small force pump, running the pressure up to the working pressure. Care must be taken, by removing delivery valves if necessary, that the pressure reaches all the suction valves.

95. The delivery valves can be tested by filling the delivery pipe or by closing the valve in the delivery pipe and pumping water into the delivery pipe between its valve and the pump delivery valve. The pump chamber must be open so that the leaks can be seen.

SURGING OF WATER IN PIPES.

96. By surging of the water flowing through pipes is meant that its velocity of flow not only is not constant, but that the direction of flow reverses for a short period. This condition often exists in pumping machinery having very long suction or delivery pipes. It may occur either in the suction pipes or in the delivery pipes, being, however, most severe in the latter. Crank-and-flywheel pumps, owing to the variation in the piston speed between the beginning and end of the stroke, are particularly liable to cause surging, which is due entirely to an irregular delivery.

97. Duplex direct-acting pumps, owing to the uniformity of delivery and the absence of heavy weights, such as flywheels, are little liable to cause surging, and when liquids must be moved through long mains, an instance of which are the long oil pipe lines, this pump is chosen. Crank-and-flywheel pumps forcing water through very high delivery pipes, as occurs in mine work, are seriously affected by the surging of the water. Air chambers do not help matters, but probably aggravate them by forming an elastic cushion for the column of water to rebound from. The effect of surging water is to vary the pressure on the pump and mains, sometimes from zero to twice the pressure due to the vertical height, resulting in broken pump chambers,

pipes, and not infrequently in damage to the working parts of the pump, for the actual resistance to these shocks is not met until they arrive at the flywheel rim.

98. The remedying of surging is not easy of attainment. Air chambers placed along the delivery pipe at intervals are employed occasionally, the aim being to break up the vibrations of the surging water and get them out of step or out of harmony with the motion of the pump. Alleviators are sometimes used in place of air chambers to relieve the shock, and not being so elastic do not encourage surging to the extent that air chambers do. When for economical reasons it is desired to use the crank-and-flywheel pump, the variations in pressure and the liability to surging can be very much reduced by using the three-throw crank with the pins set at 120° from one another.

99. Surging in long suction pipes is liable to occur especially when the water flows to the pump by gravity; this is not so difficult to overcome or so serious in its effects as surging in the delivery pipe, for the reason that the direction of the force resulting from the surge is through the pump valves and into the delivery, or in the natural direction of the water, while the shock due to surging in the delivery pipe comes against the valves and must be withstood by the machinery.

100. To prevent shocks due to surging reaching the machinery, a liberal sized air chamber is needed on the suction main near the pump, and in addition spring-loaded relief valves may also be fitted to the main. These relief valves simply limit the pressure due to an unusually heavy surge that cannot be taken care of by the air chamber.

PUMPING A MIXTURE OF WATER AND AIR.

101. In mine and artesian-well work, large quantities of air are often mixed with the water, due to local disturbances in the source of supply, such as water discharging into it in the form of spray. When such a mixture of air and water

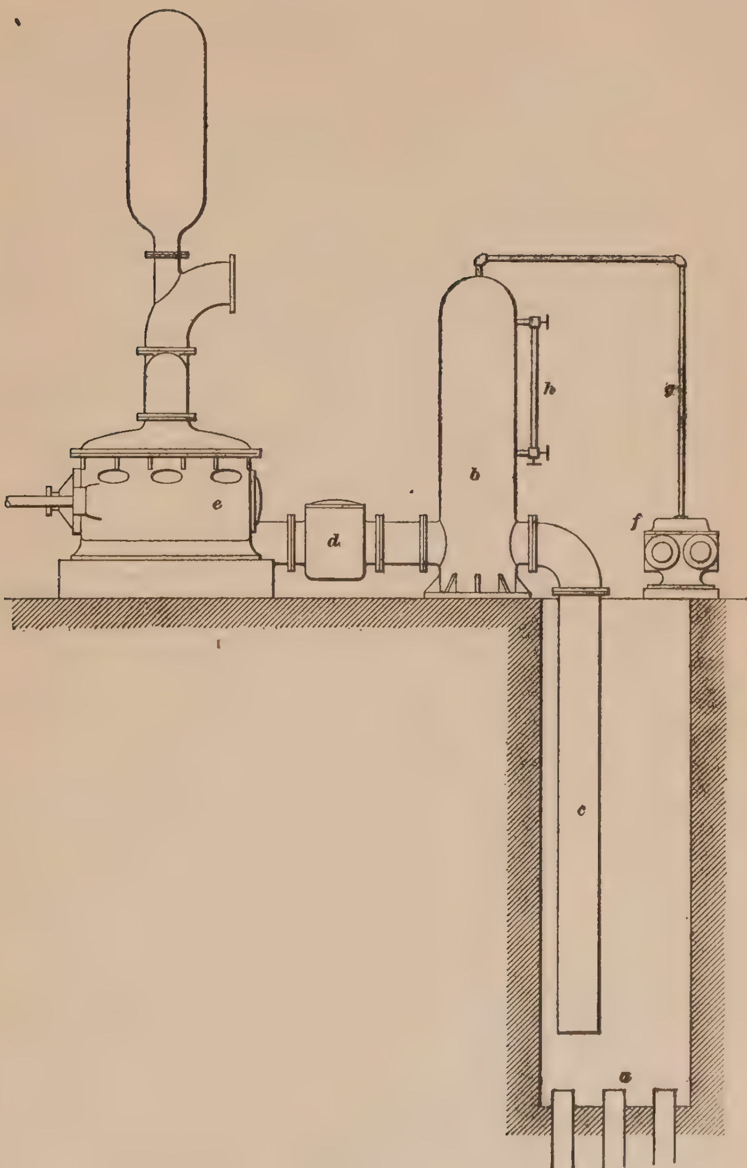


FIG. 24

is pumped, the pump will have a jerky motion, that is, instead of moving steadily it will move in jumps, and in the case of direct-acting pumps there is danger of striking the cylinder heads. Besides, on account of the uneven discharge there will be violent disturbances in the delivery pipe. The only effectual remedy is to remove the air before it arrives at the pump.

102. Fig. 24 shows the installation of a pump taking its water from an artesian well *a*, the water being highly charged with air and gas. A large suction air chamber *b* is put into the suction pipe *c*; the water passes through the strainer *d* to the pump *e*. A vacuum pump *f* is connected by the pipe *g* to the top of the air chamber and not only maintains a vacuum in the chamber, but draws the air and gas out of the water in the chamber and before it reaches the pump. The gauge glass *h* not only shows the height of water in the air chamber, but also allows the bubbles of air and gas rising through the water to be seen. The vacuum pump is simply an ordinary steam pump pumping air instead of water; it is running constantly and its speed is regulated to suit the height of the water in the air chamber.

SETTING THE VALVES OF DUPLEX STEAM PUMPS.

103. The steam valves of duplex pumps have no outside or inside lap, consequently when in their central position they just cover the steam ports leading to opposite ends of the cylinders. With all these valves a certain amount of lost motion is provided between the jam nuts and the valve. This lost motion in small pumps is within the steam chest, while in large pumps it is outside and may be adjusted while the pump is in motion. The first move in the process of setting the valves of duplex pumps is to remove the steam-chest bonnets and to place the pistons in their mid-stroke position. To do this, open the drip cocks and move each piston by prying on the crosshead, but never on the valve lever, until it comes into contact with the cylinder head.

Make a mark on the piston rod at the steam-end stuffingbox gland. Move each piston back until it strikes the opposite head, and then make a second mark on the piston rod. Half way between the first and second mark make a third one. Then, if each piston is again moved until the last mark coincides with the face of the gland, the pistons will be exactly at their mid-stroke position. After placing the pistons in their mid-position, set the valves central over the ports. Adjust the locknuts so as to allow about $\frac{3}{16}$ inch lost motion on each side. The best way of testing the equal division of the lost motion is to move each valve each way until it strikes the nut or nuts and see if the port openings are equal. When the port opening has been equalized, the valves are set. The valve motion need not be and should not be disturbed while setting the valves. Too much lost motion will tend to lengthen the stroke and may cause the piston to strike the cylinder heads, while on the other hand when there is not enough lost motion, the stroke will be perceptibly shortened. The proper amount of lost motion to give a certain length of stroke can only be found by trial for each particular pump.

104. If only one valve of a duplex pump is to be set, bear in mind that it is operated by the piston of the opposite pump. Place that piston in its mid-position and then set the valve as previously explained.

PUMPS.

(PART 3.)

CALCULATIONS RELATING TO PUMPS.

DISPLACEMENT.

1. The **displacement** of a pump for a single stroke is the volume of water that would be displaced (that is, driven out of the cylinder) by the piston or plunger during that stroke.

In calculating the displacement of a pump in a given time, care must be taken to consider the number of strokes during which water is discharged. Thus, for a single-acting pump, water is discharged only when the piston moves in one direction; and with the double-acting pump the number of strokes during which discharge occurs is equal to the total number of strokes that the piston makes. With a duplex double-acting pump, it is customary when giving the number of strokes per minute to refer only to the number of strokes made by one piston, which, obviously, is only one-half the total number of strokes made. As practice varies, however, among engineers in this respect, it is best to find out in each case, by inquiry, whether the number of strokes of one piston or of both pistons in a given time is meant when the number of strokes is given. In the case of a crank-driven pump, for a single single-acting pump the strokes will be equal to the revolutions of the crank; for a single

double-acting and a double single-acting crank-driven pump the strokes will equal twice the number of revolutions; for a triplex single-acting crank-driven pump the strokes will equal three times the number of revolutions; and for a triplex double-acting pump, six times the number of revolutions.

2. The displacement of a pump in a minute in cubic feet, gallons, or pounds is given by the following rule:

Rule 1.—*Multiply the length of stroke in inches by the mean effective area of the pump piston or plunger in square inches and the number of strokes per minute. The product is the displacement in cubic inches. To reduce the displacement to pounds, multiply by the weight of a cubic inch of the liquid pumped; to reduce to cubic feet, divide the displacement by 1,728; to reduce to Winchester gallons, divide the displacement by 231; to reduce to English imperial gallons, divide the displacement by 277.27.*

Or,

$$D_p = L\,A\,N\,S,$$

$$D_c = \frac{L\,A\,N}{1{,}728},$$

$$D_{ag} = \frac{L\,A\,N}{231},$$

$$D_{eg} = \frac{L\,A\,N}{277.27},$$

where

L = length of stroke in inches;
A = area of piston or plunger in square inches;
N = number of delivery strokes per minute;
S = weight in pounds of a cubic inch of the liquid,
D_p = displacement in pounds per minute;
D_c = displacement in cubic feet per minute;
D_{ag} = displacement in Winchester gallons per minute;
D_{eg} = displacement in English imperial gallons per minute.

3. Attention is here called to the fact that there are three different gallons in use, of which the Winchester, or wine, gallon, measuring 231 cubic inches, is most commonly

used in America. In Great Britain and her colonies the imperial gallon, holding 277.27 cubic inches, is largely used as a measure. In most English-speaking countries the beer or ale gallon of 282 cubic inches capacity is also used, but almost exclusively for measuring the liquids mentioned. When the discharge of a pump is given in gallons in the United States of America, it is always understood, unless distinctly stated otherwise, to be in gallons measuring 231 cubic inches.

4. The **mean effective area** of the piston or plunger is equal to the area corresponding to the diameter only in case of outside-packed plunger pumps. In case of inside-packed and center-packed plunger pumps and double-acting piston pumps, the mean effective area is found by dividing the sum of the piston or plunger area and the same area diminished by the area of the piston rod by 2. Thus, in a double-acting inside-packed plunger pump having a plunger 10 inches in diameter and a 2-inch piston rod, the mean effective area is $\frac{10^2 \times .7854 + (10^2 \times .7854 - 2^2 \times .7854)}{2}$ $= 76.97$ square inches. In case of a single-acting piston pump, which generally is a lift pump, the effective area will be the piston area; this should not be diminished by the area of the piston rod, although it is on the delivery side. In case of a differential pump having the plunger areas in the ratio of 1 to 2, the area of the smaller plunger is the effective area. In rough, approximate calculations of displacement, the correction for the area of the piston rod or plunger rod need not be made, and then the area of the piston or plunger is considered as the effective area. When the displacement requires to be accurately known, however, the mean effective area should be used.

EXAMPLE 1.—A single-acting plunger pump is driven by a crank whose radius is 8 inches and whose number of revolutions is 30 per minute. If the plunger is 6 inches in diameter, what is the displacement in cubic feet per minute?

SOLUTION.—The number of discharging strokes of the plunger is equal to the number of revolutions of the crank, or 30 per minute; the

length of the stroke is $8 \times 2 = 16$ inches. The area of the plunger is $6^2 \times .7854 = 28.27$ square inches. Applying rule **1**, we have

$$D_c = \frac{16 \times 28.27 \times 30}{1,728} = 7.85 \text{ cu. ft. per min.} \quad \text{Ans.}$$

EXAMPLE 2.—A center-packed double-acting duplex pump has plungers 24 inches diameter with 4-inch plunger rods. Each plunger makes 30 strokes per minute, the length of stroke being 32 inches. What is the displacement in American (Winchester) gallons per minute?

SOLUTION.—The mean effective area of the plungers is

$$\frac{24^2 \times .7854 + (24^2 \times .7854 - 4^2 \times .7854)}{2} = 446.1 \text{ square inches.}$$

Since the pump is duplex, there are $30 \times 2 = 60$ strokes per minute. Applying rule **1**, we get

$$D_{ag} = \frac{32 \times 446.1 \times 60}{231} = 3,707.8 \text{ gal. per min.,} \quad \text{Ans.}$$

DISCHARGE.

5. The **theoretical discharge** of a pump is equal to the *displacement.*

6. The **actual discharge** is generally less than the displacement, owing to leakage past the valves and piston and also to the return of water through the valves while they are in the act of closing.

SLIP.

7. The difference between the displacement and the actual discharge, expressed as a percentage of the displacement, is called the **slip** of a pump.

8. Negative Slip.—When the column of water in the suction and discharge pipes of a pump is long and the lift moderate, the energy imparted by the piston during the discharge stroke may be sufficient to keep the column in motion during all or a part of the return stroke. Under these conditions, the actual discharge may be greater than the displacement, and the slip is then said to be *negative.*

Rule 2.—*To calculate the slip of a pump, find the difference between the displacement and the actual discharge, multiply it by 100, and divide the product by the displacement. The quotient will be the slip expressed in per cent. of the displacement.*

EXAMPLE.—A single-acting plunger pump with a plunger 8 inches in diameter and 36 inches stroke discharges 33.5 cubic feet of water per minute when making 35 discharging strokes. What is the slip?

SOLUTION.—By rule **1**, the displacement is

$$\frac{36 \times 8^2 \times .7854 \times 35}{1,728} = 36.652 \text{ cubic feet per minute.}$$

By rule **2**, the slip is

$$\frac{(36.652 - 33.5) \times 100}{36.652} = 8.6 \text{ per cent., nearly. Ans.}$$

WORK DONE BY A PUMP.

9. The **useful work** in foot-pounds done by a pump is the product of the water raised in pounds multiplied by the vertical distance in feet from the surface of the water in the well or supply reservoir to the outflow end of the discharge pipe.

10. The **actual work** is always greater than the useful work. Force is required to overcome the friction of the piston or plunger in the cylinder or stuffingbox, and considerable force is also required to overcome the friction of the water in its passage through the pipes and the valves and passages of the pump. Some force must also be expended in giving the water the velocity it has when it leaves the discharge pipe.

The theoretical force required to drive the piston is equal to its area multiplied by the pressure due to a head equal to the vertical distance from the surface of the water in the well to the outlet of the discharge pipe. The actual force can be found by the aid of a pressure gauge or indicator attached to the pump cylinder, which will give the actual pressure on the piston in pounds per square inch.

According to the principles of hydraulics and the flow of water through pipes, it is evident that the power required to overcome the frictional resistance of the water will be reduced by making the pipes large and direct and the passages through the valves and pump of ample size and as direct as possible, so as to avoid loss from sudden change of direction of flow.

HORSEPOWER OF PUMPS.

11. The indicated horsepower developed in the cylinder or cylinders of a steam-driven or compressed-air-driven pump is found in exactly the same manner as with a steam engine and from the same data. The horsepower usefully expended is given by dividing the useful work done by the pump in 1 minute by 33,000. The ratio of the usefully expended horsepower to the indicated horsepower is an indication of the mechanical efficiency of the pumping apparatus considered as a whole.

12. It is often required to estimate what horsepower will be required to pump a given quantity of water per minute to a given elevation or against a given pressure. This problem can only be solved approximately by a general rule, there being a number of variable factors entering into the solution, such as the general run and length of the piping, the design of the water end, the degree of workmanship, etc. The influence of some of these factors cannot be determined beforehand with any great degree of accuracy, and for that reason any general rule for estimating the required horsepower must be based on a low mechanical efficiency of the pumping apparatus in order to leave an ample margin for safety.

13. In estimating upon the probable horsepower, it is occasionally necessary to convert a given pressure into a head of water in feet that will exert the same pressure. This can be readily done by multiplying the given pressure by 2.3.

14. If the volume of water to be discharged per minute is given in cubic feet and the vertical height from the suction level to the discharge level in feet is known, the foot-pounds of work to be done is 62.5 × volume × vertical height, taking the weight of a cubic foot of water as 62.5 pounds. Consequently, the theoretical horsepower is

$$\frac{62.5 \times \text{volume} \times \text{vertical height}}{33{,}000},$$

or

$$\frac{\text{foot-pounds of work to be done}}{33{,}000}.$$

Assuming an efficiency of 70 per cent., the actual horsepower will be

$$\frac{100 \times \text{foot-pounds of work to be done}}{70 \times 33{,}000}$$

$$= \frac{\text{foot-pounds of work to be done}}{23{,}100}.$$

Hence the following rule:

Rule 3.—*To estimate the probable horsepower required to drive a pump, multiply the weight to be discharged per minute by the vertical lift and divide by 23,100.*

Or,

$$H_e = \frac{W L}{23{,}100},$$

where H_e = estimated horsepower;
W = weight of water discharged per minute in pounds;
L = vertical lift in feet.

EXAMPLE.—About what horsepower will be required to discharge 350 gallons of water per minute, the total lift being 320 feet?

SOLUTION.—The weight of the Winchester, or ordinary American, gallon is 8.34 pounds, nearly. Hence, the weight of water to be pumped per minute is 350 × 8.34 = 2,919 pounds.

Applying rule **3**, we get

$$H_e = \frac{2{,}919 \times 320}{23{,}100} = 40 \text{ H. P., about.} \quad \text{Ans.}$$

15. When the weight of water to be discharged per minute and the pressure against which it is to be pumped are

known, the foot-pounds of work to be done is weight × pressure × 2.3. Assuming an efficiency of 70 per cent., the actual horsepower required is

$$\frac{100 \times \text{weight} \times \text{pressure} \times 2.3}{70 \times 33{,}000} = \frac{\text{weight} \times \text{pressure}}{10{,}043}.$$

Rule 4.—*To estimate the probable horsepower, multiply the weight of water to be pumped per minute by the pressure pumped against and divide by 10,043.*

Or, $$H_e = \frac{WP}{10{,}043},$$

where P = pressure per square inch;
W = weight of water per minute.

In rules **5**, **6**, **7**, and **8** the letters have the same meaning as in rules **3** and **4**.

EXAMPLE.—A pump is to pump 400 cubic feet of water per hour against a pressure of 90 pounds per square inch. Estimate the probable horsepower required.

SOLUTION.—Reducing the volume per hour to pounds per minute, we have

$$\frac{400 \times 62.5}{60} = 416.7, \text{ say } 417 \text{ pounds.}$$

Applying rule **4**, we get

$$H_e = \frac{417 \times 90}{10{,}043} = 3.7 \text{ H. P., about.} \quad \text{Ans.}$$

16. **Rule 5.**—*To estimate the vertical lift with a given horsepower, multiply the horsepower by 23,100 and divide by the weight of water to be delivered per minute.*

Or, $$L = \frac{23{,}100\,H_e}{W}.$$

EXAMPLE.—A pump driven by a 10-horsepower engine is to discharge 2,000 pounds of water per minute. How high may this water be lifted, approximately?

SOLUTION.—Applying rule **5**, we get

$$L = \frac{23{,}100 \times 10}{2{,}000} = 115.5 \text{ ft.} \quad \text{Ans.}$$

17. Rule 6.—*To estimate the probable discharge in pounds per minute, divide 23,100 times the horsepower by the vertical lift in feet.*

Or, $$W = \frac{23,100\, H_e}{L}.$$

EXAMPLE.—How many pounds of water per minute, approximately, can a pump driven by a 25-horsepower engine discharge at a height of 42 feet?

SOLUTION.—Applying rule **6**, we get

$$W = \frac{23,100 \times 25}{42} = 13,750 \text{ lb., about. Ans.}$$

18. Rule 7.—*To estimate the pressure that can be pumped against, multiply the horsepower by 10,043 and divide by the weight to be pumped per minute.*

Or, $$P = \frac{10,043\, H_e}{W}.$$

EXAMPLE.—A 9-horsepower pump is to discharge 6,000 pounds of water per minute. Estimate against what pressure this can be discharged.

SOLUTION.—Applying rule **7**, we get

$$P = \frac{10,043 \times 9}{6,000} = 15 \text{ lb. per sq. in. Ans.}$$

19. Rule 8.—*To estimate the probable discharge in pounds per minute, multiply the horsepower by 10,043 and divide by the pressure to be pumped against.*

Or, $$W = \frac{10,043\, H_e}{P}.$$

EXAMPLE.—How much water may a pump be estimated to discharge in Winchester gallons per minute when the pump is 40-horsepower and pumps against a pressure of 100 pounds per square inch?

SOLUTION.—Applying rule **8**, we get

$$W = \frac{10,043 \times 40}{100} = 4,017.2 \text{ pounds per minute.}$$

Since a Winchester gallon weighs 8.34 pounds, we have

$$\frac{4,017.2}{8.34} = 481.7 \text{ gal. per min. Ans.}$$

SIZE OF PISTONS AND PLUNGERS.

20. Before the size of a piston or plunger for the water end of a pump can be determined, the quantity of water to be pumped and the piston speed must be known. The piston speed is the number of feet traveled per minute by the plunger when *discharging* water; that is, it equals the length of the stroke in feet multiplied by the number of *working* strokes per minute. If the pump is double-acting, the number of working strokes is the same as the total number of plunger strokes, both forward and back; if single-acting, half that number. If the pump is duplex, it is advisable to consider only one side in determining the size of plunger or piston, designing it to suit one-half the total quantity of water to be delivered. In direct-acting steam pumps the piston speed is generally about 100 feet; at least it is customary to design them on this assumption, and then to run the pump faster or slower to suit the required delivery, opening or closing the throttle valve to vary the speed of the pump.

21. Knowing the actual volume of water to be discharged in 1 minute in cubic feet, the plunger or piston area in square feet will be $\frac{\text{discharge}}{\text{piston speed}}$, theoretically. But in practice the diameter of the plunger or piston is given in inches, hence the area should be expressed in square inches.

Then, $$\text{area} = \frac{\text{discharge in cubic feet} \times 144}{\text{piston speed in feet}},$$

and the corresponding diameter in inches will be

$$\sqrt{\frac{\text{discharge} \times 144}{.7854 \times \text{piston speed}}}.$$

22. Since there is always more or less slip of the water, it is usual to design the pump on the assumption that it must pump 1.25 times the actual amount of water. On this assumption the plunger or piston diameter in inches will be

$$\sqrt{\frac{\text{discharge} \times 1.25 \times 144}{.7854 \times \text{piston speed}}},$$

or $$\sqrt{\frac{229 \times \text{discharge}}{\text{piston speed}}}.$$

Rule 9.—*To find the diameter of a plunger or piston in inches, multiply the discharge in cubic feet per minute by 229 and divide the product by the piston speed in feet per minute. Extract the square root of the quotient.*

Or, $$d = \sqrt{\frac{229\,D}{S}},$$

where d = diameter of piston or plunger in inches;
D = actual discharge in cubic feet per minute;
S = piston speed.

When the discharge is given in pounds, gallons, or any other unit of volume, it should be reduced to cubic feet before applying rule **9.**

EXAMPLE.—What should be the diameter of a pump plunger required to discharge 130 Winchester gallons per minute, the speed of the plunger being 90 feet per minute?

SOLUTION.—Reducing the gallons to cubic feet, we have

$$\frac{130 \times 231}{1{,}728} = 17.378 \text{ cubic feet per minute.}$$

Applying rule **9,** we get

$$d = \sqrt{\frac{229 \times 17.378}{90}} = 6.65 \text{ in., nearly.} \quad \text{Ans.}$$

23. Rule 10.—*To estimate the probable discharge in cubic feet, square the diameter of the plunger or piston in inches and multiply by the piston speed. Divide the product by 229.*

Or, $$D = \frac{d^2\,S}{229}.$$

EXAMPLE.—How many pounds of water per hour may a duplex double-acting pump be expected to discharge when the diameter of the plungers is 6 inches, the length of stroke 24 inches, and each plunger makes 40 strokes per minute?

SOLUTION.—The piston speed is $\frac{24}{12} \times 40 = 80$ feet per minute. The probable discharge per minute in cubic feet, by rule **10,** is

$$D = \frac{6^2 \times 80}{229};$$

per hour, $$D = \frac{6^2 \times 80 \times 60}{229},$$

The discharge in pounds per hour, taking 62.5 pounds as the weight of a cubic foot of water, is

$$D = \frac{6^2 \times 80 \times 60 \times 62.5}{229}$$

for one side of the pump. For both sides,

$$D = \frac{6^2 \times 80 \times 60 \times 62.5 \times 2}{229} = 94,323 \text{ lb.} \quad \text{Ans.}$$

In applying rule **10** it is to be observed that the result will be less than given by multiplying the displacement per stroke by the number of strokes per minute, as called for by rule **1**. The reason for this discrepancy is obvious; rule **1** gives the theoretical discharge, while rule **10** gives about what the pump may actually be expected to discharge.

24. In direct-acting steam pumps the normal piston speed is generally 100 feet per minute. On this basis the probable discharge in cubic feet, by rule **10**, is $D = \frac{d^2 \times 100}{229}$, and in Winchester gallons the discharge is $\frac{d^2 \times 100 \times 1,728}{231 \times 229} = 3.26\, d^2$.

Rule 11.—*To roughly approximate the probable normal discharge of a direct-acting steam pump in gallons, multiply the square of the diameter of the plunger or piston by 3.26.*

Or, $$D_g = 3.26\, d^2,$$

where D_g = discharge in gallons per minute;
d = diameter of piston or plunger in inches.

25. The theoretical normal discharge in gallons per minute at a piston speed of 100 feet is given almost exactly by multiplying the square of the diameter of the plunger or piston by 4. For a duplex pump the discharge is double that given by rule **11**.

EXAMPLE.—What may the discharge in gallons of a duplex pump with 6-inch plungers be roughly estimated at?

SOLUTION.—Applying rule **11**, we get $D_g = 3.26 \times 6^2$ for each side, or

$$D_g = 3.26 \times 6^2 \times 2 = 235 \text{ gal. per min.} \quad \text{Ans.}$$

26. Having determined the proper plunger or piston diameter for the chosen piston speed, it remains to choose either a length of stroke or a number of strokes in order to determine either the number of strokes or the length of stroke. The ratio of the diameter to the length of stroke varies between very wide limits in practice, being as low as 1 : 1 and as high as 1 : 5. Obviously, the greater the ratio, the fewer times will the valves have to be moved, hence a great ratio is generally chosen for pumps that have to run continuously in a hard, rough service. Having chosen a length of stroke, use the following rule:

Rule 12.—*To find the number of strokes, divide the piston speed in feet by the chosen length of stroke in feet. To find the length of stroke in feet, divide the piston speed in feet by the number of delivery strokes per minute.*

Or,
$$N = \frac{P}{L},$$

and
$$L = \frac{P}{N},$$

where P = piston speed;
N = number of delivery strokes per minute;
L = length of stroke in feet.

EXAMPLE.—What should be the length of stroke for a piston speed of 100 feet if the number of strokes per minute is 40?

SOLUTION.—Applying rule **12**, we get

$$L = \tfrac{100}{40} = 2.5 \text{ ft.},$$

or
$$2.5 \times 12 = 30 \text{ in. Ans.}$$

SIZE OF STEAM END.

27. In a direct-acting steam pump the size of the steam-end cylinder depends on two factors, which are the steam pressure available and the resistance against which the pump is to force the water. The stroke of the steam piston and water piston obviously are the same, both being rigidly connected to the same rod.

28. The forces acting on the steam piston and water piston are equal when the area of the steam piston × the steam pressure = area of water piston × pressure pumped against. But in order that there may be an ample margin to overcome the frictional resistances, which make the actual resistance to the motion of the water piston greater and lessen the force that impels the steam piston forwards, the area of the steam piston should be, at least, 40 per cent. in excess of its theoretical area. On this basis, we have area of steam piston

$$= \frac{1.4 \times \text{area of water piston} \times \text{pressure}}{\text{steam pressure}},$$

and diameter of steam piston

$$= \sqrt{\frac{1.4 \times \text{area of water piston} \times \text{pressure}}{.7854 \times \text{steam pressure}}},$$

or diameter of steam piston

$$= \sqrt{\frac{1.8 \times \text{area of water piston} \times \text{pressure}}{\text{steam pressure}}}.$$

Rule 13.—*To find the minimum diameter of the steam piston of a direct-acting steam pump, multiply 1.8 times the area of the water piston in square inches by the pressure in pounds per square inch to be pumped against; divide by the available steam pressure and extract the square root of the quotient.*

Or, $$d_m = \sqrt{\frac{1.8\,a\,p}{P}},$$

where d_m = minimum diameter of steam piston in inches;
a = area of water piston;
p = pressure to be pumped against;
P = steam pressure available.

Example.—What should be the minimum diameter of the steam piston for a pump having a plunger 8 inches in diameter, the available steam pressure being 75 pounds per square inch and the water to be pumped against a pressure of 200 pounds per square inch?

Solution.—The area of the plunger is $8^2 \times .7854 = 50.27$ square inches. Applying now rule **13**, we get

$$d_m = \sqrt{\frac{1.8 \times 50.27 \times 200}{75}} = 15.5 \text{ in.} \quad \text{Ans.}$$

29. It is to be observed that rule **13** applies equally well to steam- and air-driven pumps. It can also be applied to simple pumps of the crank-and-flywheel type using steam expansively. In the latter case, the mean effective pressure throughout the stroke must be taken as the available steam pressure. Rule **13** is especially useful in deciding whether a given pump will pump against a known pressure with the existing sizes of steam and water pistons. It will also be found very useful in selecting a pump for a given service from the catalogues of manufacturers.

30. In boiler-feed pumps the steam pressure available and the pressure pumped against are practically equal, so that it might be expected that the area of the steam piston would be made about 40 per cent. larger than the area of the water piston. In actual practice it is found, however, that pump manufacturers prefer to make the steam piston about 3 times the area of the water piston in very small pumps and about twice the area of the water piston in large pumps. The steam piston of boiler-feed pumps is made so largely in excess of what it really needs to be merely as a matter of safety; its large size simply tends to insure a prompt starting of the pump under almost all conditions likely to arise in practice.

31. The steam end of direct-acting pumps and of direct-connected crank-and-flywheel pumps, where the steam and water pistons move together, is rarely proportioned on the basis of horsepower required to do the work, it being much easier to calculate the size of the steam end by rule **13**.

32. When a power pump is driven by a separate steam engine, through the intervention of belting or gearing, the engine itself is generally selected on the basis of horsepower

required to do the work, and then the question as to what size of engine to use presents itself. This problem is capable of an infinite number of solutions, since a variation of either of the two factors—piston speed and mean effective pressure—will cause a difference in size. In general, the steam engine for driving a pump is selected in exactly the same manner, as far as its size is concerned, as a steam engine for any other service.

THE DUTY OF STEAM PUMPS.

DEFINITION.

33. The ratio between the work done by a pump and a certain amount of coal, steam, or heat units used to do the work is called the **duty** of the pump.

During a certain time, say an hour or a day, the pump will raise a quantity of water through a certain height and thus perform a definite amount of work. To do this work, the pump has received from the boilers a certain number of heat units or a number of pounds of steam; or, if the boilers are included as a part of the system, the work has been accomplished by consuming a certain amount of coal. The pump is credited with the work it has performed in the stated time and is charged with the number of heat units, pounds of steam, or pounds of coal it has used in doing the work. It is plain that the economy of the pump or pumping engine is measured by the ratio of the work performed to the steam consumed or the coal burned. Thus, if one pump does 50,000,000 foot-pounds of work with a coal consumption of 100 pounds and another under the same conditions does 36,000,000 foot-pounds and consumes only 60 pounds of coal, the latter is evidently the more economical, since the ratio of work to coal consumption is larger, being $(36,000,000 \div 60) \times 100 = 60,000,000$ foot-pounds of work with a coal consumption of 100 pounds.

DUTY BASED ON COAL CONSUMPTION.

34. When the duty is based on the consumption of coal, it is customary to assume 100 pounds of coal as the fuel unit; that is, the duty is defined as the number of foot-pounds of work performed for each 100 pounds of coal burned. Then,

$$\text{Duty} = \text{foot-pounds of work} \div \frac{\text{pounds of coal}}{100},$$

or,
$$\text{Duty} = \frac{\text{foot-pounds of work} \times 100}{\text{pounds of coal}}.$$

Rule 14.—*To find the duty of a pump per 100 pounds of coal, multiply together 100, the weight of water pumped in a given time in pounds, and the vertical distance in feet from the level of supply to the level of discharge. Divide the product by the coal consumption in the same time in pounds.*

Or,
$$D = \frac{100\, w\, h}{W},$$

where
D = duty;
w = weight of water in pounds;
W = weight of coal in pounds;
h = vertical lift in feet.

EXAMPLE.—A pump raises 130,000 pounds of water 60 feet and the operation requires the combustion of 25 pounds of coal. What is the duty?

SOLUTION.—Applying rule **14**, we have

$$D = \frac{100 \times 130{,}000 \times 60}{25} = 31{,}200{,}000 \text{ ft.-lb. per 100 lb. of coal. Ans.}$$

35. The duty based on the coal consumption is of practical value, as it gives an idea of the coal required by a pump of a given type for the performance of a stated quantity of work. It is clear, however, that if a comparison of the merits of two pumps is to be made, the coal must be of the same quality in each case. Further, the boilers supplying steam to the pumps should be of the same type or at least have the same evaporative capacity. This is a point of great importance. One hundred pounds of good bituminous or anthracite coal may, under favorable conditions, evaporate

1,000 to 1,100 pounds of water; that is, furnish that number of pounds of steam to the pump. In many cases, however, the 100 pounds of coal, if of inferior quality and burned under a poor boiler, will not furnish the pump more than 450 to 600 pounds of steam. Under such conditions the duty of the pump based on the coal consumption would not be a fair indication of its efficiency and would not serve as a satisfactory basis for comparing it with other pumps.

DUTY BASED ON STEAM CONSUMPTION.

36. In order to avoid the drawbacks incidental to basing the duty of pump on the coal consumption, it is the custom of some pump makers to specify that the coal consumption shall be estimated on the supposition that a pound of coal evaporates 10 pounds of water, or, in other words, furnishes 10 pounds of steam to the pump. To make this clear, suppose that in a duty trial 32,000 pounds of steam were used by the pump; the duty of the pump would be calculated on the assumption that the coal consumption was $32,000 \div 10 = 3,200$ pounds, though 5,000 pounds might actually have been used. If 1 pound of coal is assumed to furnish 10 pounds of steam, 100 pounds of coal will furnish 1,000 pounds of steam; hence, the duty based on steam consumption may be defined as the number of foot-pounds of work done by the pump per 1,000 pounds of dry steam supplied it. Then,

$$\text{Duty} = \frac{\text{foot-pounds of work} \times 1,000}{\text{pounds of steam}}.$$

Rule 15.—*To find the duty of a pump per 1,000 pounds of dry steam, multiply together the weight of water pumped in pounds, the vertical distance in feet from the level of supply to the level of discharge, and 1,000. Divide by the weight of steam supplied in pounds.*

Or,
$$D = \frac{1,000\, w\, h}{S},$$

where S = weight of dry steam supplied in pounds and the other letters have the same meaning as in rule **14**.

EXAMPLE.—A pump lifted 7,920,000 pounds of water 126 feet with 8,100 pounds of steam. What is its duty?

SOLUTION.—Applying rule **15**, we get

$$D = \frac{1,000 \times 7,920,000 \times 126}{8,100}$$

$= 123,200,000$ ft.-lb. of work per 1,000 lb. of dry steam. Ans.

37. The basis of 1,000 pounds of dry steam is more scientific and better adapted for duty trials than that of 100 pounds of coal, but it is open, nevertheless, to objections. Not only is it difficult to determine the exact weight of dry steam entering the pump, but also 1,000 pounds of steam at 160 pounds pressure will do more work in the cylinder than 1,000 pounds of steam at 60 pounds pressure. If scientific accuracy is sought, the pressure of the steam should be specified in addition to the weight.

DUTY BASED ON HEAT UNITS SUPPLIED.

38. On account of the objections to the basis of comparison then used, a committee of The American Society of Mechanical Engineers in 1891 recommended a new basis for the estimation of duty. Whether the furnace consumes 100 or 200 pounds of coal, whether the steam is at 60 or 160 pounds pressure, wet or dry, the steam cylinders of the pump or pumping engine receive in a given time a definite number of British thermal units. We have seen that if each of two pumps is allowed 100 pounds of coal to do a certain amount of work, one of the pumps may be at a disadvantage on account of the poor quality of the coal or the inefficiency of the boiler. If each is allowed 1,000 pounds of dry steam, there may be an inequality because of a difference in the steam pressure in the two cases. If, however, each pump is furnished with an equal number of heat units, each has exactly the same stock in trade, and the merit of each pump can be gauged by the use it makes of the heat units furnished it, that is, by the ratio of the work performed to number of heat units supplied.

39. If a pound of water has a temperature of 212°, it requires 966.1 B. T. U. to change it to steam at atmospheric pressure. If the water has originally a lower temperature or is converted into steam at higher pressure, more B. T. U. are required to accomplish the change. Roughly speaking, if the temperature of the feed and pressure of the steam are not given, about 1,000 to 1,100 B. T. U. are equivalent to a pound of steam. Therefore, 1,000 pounds of steam are equivalent to about $1,000 \times 1,000 = 1,000,000$ B. T. U.

40. Looking at the question in another light, a pound of good coal when burned produces about 13,500 to 14,000 B. T. U. by the combustion. A boiler of fairly good efficiency will utilize perhaps 10,000 of these 13,500 B. T. U., the rest being lost by radiation, in the production of chimney draft, and in other ways. From 100 pounds of coal the boiler is able to extract $100 \times 10,000 = 1,000,000$ B. T. U., which are eventually given up to the pump. It thus appears that 100 pounds of coal and 1,000 pounds of steam are each approximately equivalent to 1,000,000 B. T. U.; for this reason, the committee of The American Society of Mechanical Engineers recommended that the new basis for estimating duty should be 1,000,000 B. T. U.

41. The heat-unit basis is now very extensively used and is recommended in preference to the others. It may be expressed as follows:

The duty of a pumping engine is equal to the total number of foot-pounds of work actually done by the pump divided by the total number of heat units in the steam used by the pump, and this quotient multiplied by 1,000,000. The heat units in the steam used for driving the auxiliary machinery, such as the air pump and circulating pump of the condenser, if one is used, and the boiler-feed pumps are charged as heat units supplied to the pump.

42. The number of foot-pounds of work done by the pump is to be found as follows: A pressure gauge is attached to the discharge pipe and a vacuum gauge to the

suction pipe, both as near the pump as convenient; then the net pressure against which the pump plunger works is equal to the sum or difference in the pressures shown by these two gauges increased by the hydrostatic pressure due to the difference in level of the points in the pipes to which they are attached. In case the gauge in the suction pipe indicates a vacuum, the sum of the pressures indicated by the gauges is taken, but when the water flows into the suction pipe under a head, so that the suction gauge indicates a pressure above the atmospheric pressure, the difference in the two pressures indicated by the gauges is taken.

43. The number of foot-pounds of work done during the trial is equal to the continued product of the net area of the plunger in square inches (making allowance for piston rods), the length of the plunger stroke in feet, the number of plunger strokes made during the trial, and the net pressure in pounds per square inch against which the plungers work.

44. The pressure corresponding to the vacuum in inches indicated by the gauge on the suction pipe is found by multiplying the gauge reading in inches by .4914, and the pressure corresponding to the difference in the level of the two gauges by multiplying this difference in feet by .434. The number of heat units furnished to the pump is the number of British thermal units in the steam from the boilers and is to be determined by an evaporation test of the boilers.

Rule 16.—*To determine the duty of a pump per 1,000,000 B. T. U., multiply the net pressure against which the plunger works, in pounds per square inch, by the net area of the plunger in square inches, by the average length of stroke in feet, the total number of delivery strokes made during the trial, and by 1,000,000. Divide the product by the total number of B. T. U. supplied during the trial.*

Or, $$D = \frac{1{,}000{,}000\,(P \pm p + S)\,A\,L\,N}{H}.$$

where D = duty;

P = pressure in pounds per square inch in the discharge pipe;

p = pressure in pounds per square inch in the suction pipe, to be added in case of a vacuum and to be subtracted in case of pressure above atmospheric pressure in the suction pipe;

S = pressure in pounds per square inch corresponding to difference in level between the gauges;

A = average effective area of plunger in square inches;

L = length of stroke of pump plunger in feet;

N = total number of delivery strokes;

H = total number of B. T. U. supplied.

EXAMPLE.—A crank-and-flywheel pump has two double-acting water plungers, each 20 inches in diameter and 36 inches stroke. Each plunger has a piston rod 3 inches in diameter extending through one pump-cylinder head.

During a 10-hour duty trial the total heat in the steam supplied to the engine was 35,752,340 B. T. U. and the engine made 9,527 revolutions. If the average pressure indicated by a gauge on the discharge pipe was $95\frac{1}{2}$ pounds, the average vacuum indicated by a gauge on the suction pipe $8\frac{1}{4}$ inches, and the difference in level between the centers of the vacuum and the pressure gauge 8 feet, what was the duty?

SOLUTION.—The area of a plunger 20 inches in diameter is 314.16 square inches and the cross-sectional area of a rod 3 inches in diameter is 7.07 square inches. Since the rod extends through only one end of the pump cylinder, the average effective area of the two ends of each plunger is $314.16 - \frac{7.07}{2} = 310.63$ square inches.

The pressure corresponding to a vacuum of $8\frac{1}{4}$ inches is $p = 8.25 \times .4914 = 4.05$ pounds per square inch, and the pressure corresponding to a difference in level of 8 feet is $S = 8 \times .434 = 3.47$ pounds per square inch.

Since there are two double-acting plungers, the total number of plunger strokes corresponding to 9,527 revolutions is $N = 9,527 \times 4 = 38,108$.

Applying rule **16**, we get

$$D = \frac{1,000,000 \times (95.5 + 4.05 + 3.47) \times 310.63 \times 3 \times 38,108}{35,752,340}$$

$$= 102,328,800 \text{ ft.-lb. per } 1,000,000 \text{ B. T. U.} \quad \text{Ans.}$$

DUTY BASED ON VOLUME OR WEIGHT PUMPED.

45. In large pumping plants it often happens that the pressure pumped against is either constant or practically so. In such a plant a record is often kept for the purpose of comparing the performance of the plant from week to week or month to month with its former performances. The records may be kept in number of gallons pumped per pound of coal; in cubic feet pumped per pound of coal; in weight of water in pounds or tons pumped per pound of coal; or the record may be kept per ton or bushel of coal, etc. Duty computed on such a basis is spoken of as **gallon duty, cubic-foot duty, pound duty, ton duty,** etc.; while such a duty is very valuable in showing variations in efficiency of a given plant at different times, it cannot be used as a basis of comparison between the performances of different pumping plants, and when so used will be utterly misleading.

Instead of keeping the records in terms of quantity of water pumped per pound of coal, they may advantageously be kept in terms of water pumped per dollar; the records then show variations in efficiency in their true light.

EXPRESSING THE DUTY OF A PUMP.

46. The question "in what terms shall the duty of a pump be expressed" depends for its answer on the purpose for which it is required that the duty be known. If the duty is merely required to be known in order that the performances of a given pump at different times may be compared with one another, the duty may be based on coal consumption, steam consumption, or volume pumped per some unit of fuel or money. If, however, the performance of a pump is to be compared with that of others working probably under entirely different conditions, the foot-pounds of work done per 1,000,000 B. T. U. is the only true basis of comparison.

AVERAGE DUTIES.

47. Small direct-acting pumps for general service have a duty of 15,000,000 foot-pounds per 1,000 pounds of steam used. Compound direct-acting pumps of 5,000,000 gallons capacity in 24 hours should give a duty of 50,000,000 foot-pounds per 1,000 pounds of steam used. Large municipal pumping engines of 20,000,000 gallons capacity in 24 hours have given a duty of 160,000,000 foot-pounds per 1,000 pounds of dry steam used by the engine.

Centrifugal and rotary pumps have a duty depending on the type of engine used to drive them, and since they usually run at high speed and the conditions for economical performance are good, an economical type of engine can be used and the duty of the combined unit thus made to compare very favorably with that of the reciprocating pump.

48. Tests of the duty of pumps and pumping engines have generally been made when the machinery was in first-class condition. It is customary to run these machines from 6 months to 1 year after they are installed before making the test, the object being to bring all the journals into a good bearing condition; also, the piston and all the other rubbing surfaces will be much improved by the polishing and the working of oil into the pores of the iron during running. These high duties can only be maintained by the closest attention to every detail by the operating engineer. Indicator cards should be taken from both the steam and the water ends of the pumps every week and closely compared with previous indications to see that the highest state of efficiency is being maintained within the working parts of the pump.

EFFICIENCY OF VARIOUS TYPES OF PUMPS.

49. When the **efficiency of a pump** is spoken of, its *mechanical* efficiency is generally meant, unless stated otherwise. This is measured by dividing the actual or net horsepower of the machine by its indicated horsepower, and the quotient, when multiplied by 100, will be the efficiency

expressed in per cent. Very small direct-acting steam pumps have an efficiency of about 50 per cent., the efficiency increasing with the size of the pump up to about 80 per cent. The efficiency of direct-acting steam pumps and also of pumps in general increases with the size by reason of the decrease in the ratio that the frictional resistances bear to the indicated horsepower as the size of the pipes and passages is increased. The reason that the frictional resistances decrease can readily be seen when it is considered that by doubling the diameter of a pipe and keeping the velocity of flow the same, the discharge will be increased four times, while the surface that the water is rubbing against is only doubled.

50. Large vertical municipal pumping engines have shown an efficiency as high as 96 per cent.; horizontal medium-size crank-and-flywheel pumps show efficiencies as high as 90 per cent. The efficiency of centrifugal, rotary, and screw pumps varies between 40 and 66 per cent., about, depending on the size; small pumps are less efficient than larger ones. This efficiency of centrifugal, rotary, and screw pumps is the *efficiency of the pump itself*, and not the combined efficiency of the pump and engine, or motor, driving it.

SIZE OF SUCTION AND DELIVERY PIPES.

51. Experience has demonstrated that for satisfactory work the flow of water in the suction pipes of pumps should not exceed 200 feet per minute, and it should not be more than 500 feet in the delivery pipe for a duplex double-acting pump, or 400 feet for a single-cylinder double-acting pump.

Knowing the volume of water that is to flow through or to be discharged from a pipe in 1 minute, the area of the suction and delivery pipes can readily be determined.

The volume of water in cubic feet discharged from a pipe in 1 minute is equal to the velocity in feet per minute times the area of the pipe in square feet. Then,

$$\text{the area of the pipe} = \frac{\text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}.$$

As there are 144 square inches in a square foot,

$$\text{the area of the pipe in square inches} = \frac{144 \times \text{volume in cubic feet per minute}}{\text{velocity in feet per minute}}.$$

Rule 17.—*To find the area of a pipe in square inches to discharge a given volume of water per minute, divide the product of the volume in cubic feet and 144 by the allowable velocity in feet per minute.*

Or, $$A = \frac{144\,V}{v},$$

where A = area of pipe in square inches;
V = volume to be discharged per minute;
v = allowable velocity.

When the weight of water is given in pounds, divide it by 62.5 to reduce it to cubic feet; when the volume is given in Winchester gallons, divide it by 7.48 to reduce it to cubic feet.

EXAMPLE.—What should be the areas of the suction and delivery pipes for a single double-acting pump that is to discharge 6,250 pounds of water per minute?

SOLUTION.—Reducing the weight to cubic feet, we have $\frac{6{,}250}{62.5} = 100$ cubic feet. Then, applying rule **17**, we have

$$A = \frac{144 \times 100}{200} = 72 \text{ square inches}$$

as the area of the suction pipe, and

$$A = \frac{144 \times 100}{400} = 36 \text{ square inches}$$

as the area of the delivery pipe. The nearest standard nominal sizes of pipe to be used would be 10-inch and 7-inch. Ans.

52. The velocity with which water will flow through the delivery pipe of a pump when the area of the water cylinder, the area of the delivery pipe, and the piston speed of the pump are known, is given by the following rule:

Rule 18.—*Multiply the area of the water piston by the piston speed and divide this product by the area of the delivery pipe.*

Or,
$$v = \frac{a\,S}{A},$$

where v = velocity in feet per minute;
A = area of delivery pipe in square inches;
a = area of water piston in square inches;
S = piston speed in feet per minute.

EXAMPLE.—If the water piston of a pump has an area of 12 square inches and moves at a speed of 100 feet per minute, what will be the velocity of the water in the delivery pipe if the latter has an area of 2 square inches?

SOLUTION.—Applying rule **18**, we get

$$v = \frac{12 \times 100}{2} = 600 \text{ ft. per min.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. The plungers of a center-packed double-acting duplex pump are 20 inches in diameter and the plunger rods are $3\frac{1}{2}$ inches in diameter. Each plunger makes 45 strokes per minute, the length of stroke being 24 inches. What is the displacement in cubic feet per minute?
Ans. 386.69 cu. ft.

2. In the above example, if the pump delivers but 360 cubic feet per minute, what is the slip? Ans. 6.9 per cent.

3. Approximately, what horsepower will be required to deliver 60 cubic feet of water per minute, the total lift being 470 feet?
Ans. 76.3 H. P.

4. What is the probable horsepower required to deliver 3,500 gallons of water per hour against a pressure of 115 pounds per square inch?
Ans. 5.57 H. P.

5. A pump driven by a 25-horsepower engine is to discharge 60 cubic feet of water per minute. How high may this water be lifted, approximately? Ans. 154 ft.

6. Approximately, how many gallons of water per hour can a pump driven by a 30-horsepower engine deliver at a height of 65 feet?
Ans. 76,701.7 gallons.

7. Approximately, against what pressure can a 20-horsepower pump discharge 2,500 cubic feet of water per hour?
Ans. 77 lb. per sq. in.

8. About how many cubic feet of water per minute may a 75-horsepower pump be expected to discharge against a pressure of 150 pounds per square inch? Ans. 80.3 cu. ft. per min.

9. A pump is required to discharge 1,800 cubic feet of water per hour. If the speed of the plunger is 100 feet per minute, what should be the diameter of the plunger? Ans. 8.29 in., nearly.

10. If the plunger of a double-acting pump is 10 inches in diameter and the length of stroke is 24 inches, how many gallons of water per hour may the pump be expected to deliver if it makes 45 strokes per minute? Ans. 17,639 gal. per hr.

11. Roughly estimate the discharge in gallons of a direct-acting steam pump having a plunger 7 inches in diameter.

Ans. 159.7 gal. per min.

12. If the piston speed is 90 feet per minute and the length of stroke 2 feet, how many strokes per minute will the pump make? Ans. 45.

13. Calculate the minimum diameter of the steam piston for a pump having a plunger 12 inches in diameter, the pressure to be pumped against being 175 pounds per square inch and the available steam pressure 100 pounds per square inch. Ans. 18.87 in.

14. What is the duty per 100 pounds of coal of a pump that raises 330,000 pounds of water 125 feet and requires 110 pounds of coal to perform the operation? Ans. 37,500,000 ft.-lb.

15. If 20,016 pounds of steam are consumed by a pump in lifting 1,200,000 gallons of water 150 feet, what is the duty per 1,000 pounds of dry steam? Ans. 75,000,000 ft.-lb.

16. A double-acting pump has a stroke of 40 inches; the diameter of the plunger is 24 inches and the diameter of the piston rod, which extends through one pump-cylinder head, is $3\frac{1}{4}$ inches. During a 12-hour duty trial the total heat supplied to the engine was 47,652,500 B. T. U. and the engine made 23,200 strokes. What was the duty of the pump per 1,000,000 B. T. U. if the average pressure indicated by the gauge on the discharge pipe was 122 pounds, the average vacuum indicated by a gauge on the suction pipe 5 inches, and the difference in level between the centers of the vacuum gauge and pressure gauge was 10 feet? Ans. 93,555,123 ft.-lb.

17. Calculate the area of the suction and delivery pipes for a single-acting pump that is to discharge 1,250 gallons of water per minute.

Ans. $\begin{cases} \text{Suction pipe, 120.3 sq. in.} \\ \text{Delivery pipe, 60.15 sq. in.} \end{cases}$

18. If the plunger of a single-acting pump has a speed of 85 feet per minute and a diameter of 6 inches, what will be the velocity of the water in the delivery pipe if the latter has an area of 6 square inches? Ans. 400.5 ft. per min.

SELECTION OF PUMPS.

SERVICE OF DIFFERENT TYPES OF PUMPS.

53. Introduction.—The service for which a pump is required determines its general type, that is, whether it is to be a plunger pump, a rotary pump, a centrifugal pump, or a screw pump.

54. Reciprocating Pumps.—The various types of reciprocating pumps are selected when high efficiency is required and a fluid for which they are suited is to be pumped.

55. Rotary Pumps.—The rotary pump is chosen when the fluid to be pumped is water holding in suspension large masses of soft material. It is much used in paper mills for pumping the pulp from one stage of its manufacture to another. Rotary pumps are small and occupy, relatively, but little space for their capacity; they are also light, simple, and inexpensive, but are low in efficiency and are short lived, particularly if the material pumped contains much sand or other grit. The rotary pump is used with good success on some steam fire-engines, where light weight and simplicity are more important than high efficiency.

56. Centrifugal Pumps.—Centrifugal pumps are used where large volumes of water are to be lifted to moderate heights. They are also well adapted for pumping large quantities of dirty water, and, hence, are also much used for dredging and for sewage pumping. The efficiency of the centrifugal pump is low, but it is extremely simple and occupies comparatively little space for its capacity. Like the rotary pump, it has no valves and the flow is continuous. It is less affected by sand and grit than is the rotary pump. Neither the rotary pump nor the centrifugal pump requires much, if any, foundation.

57. Displacement Pumps.—Under the head of displacement pumps may be classed the pulsometer, which has

no running parts. This type of pump is well adapted for pumping all kinds of gritty water and is used for sinking and contractor's purposes. It is very simple in construction, low in first cost, and is not liable to get out of order. The class of pumps known as air lifts are principally used for artesian-well service; they require an air compressor for operation, but the apparatus itself is simple and low in first cost.

58. Screw Pumps.—Screw pumps are adapted for the handling of thick liquids, such as hot tar, pitch, paraffin, soap, etc. They have a uniform discharge and occupy small space; a much higher efficiency is claimed for them than for rotary or centrifugal pumps.

RECIPROCATING PUMPS.

CLASSIFICATION.

59. The reciprocating pump is, in general, the most efficient and hence the most common pump. It is built in a large variety of designs to suit different conditions and varies in size between very wide limits. Reciprocating pumps may be classified in accordance with the service for which they are intended as boiler-feed pumps, general-service pumps, tank or light-service pumps, fire pumps, low steam-pressure pumps, pressure pumps, mine pumps, sinking pumps, ballast pumps, wrecking pumps, deep-well pumps, sewage pumps, vacuum pumps, power pumps, municipal pumping engines, etc.

BOILER-FEED PUMPS.

60. Boiler-feed pumps are used for supplying steam boilers with their necessary water supply. For low pressures they are usually made of the piston pattern or the inside-packed plunger patterns. The cylinders are generally brass lined; the valves are brass or hard composition, with

composition springs and guards, and the pumps, hence, are suitable for handling hot water. For pressures above 135 pounds the outside-packed plunger type is preferred. Boiler-feed pumps are made both vertical and horizontal and for pressures from 50 pounds to 300 pounds per square inch. They vary in size from those having water plungers 1 inch in diameter to those having plungers 10 inches in diameter. The single-cylinder type is much used for boiler feeding, but, perhaps undeservedly, they have not the reputation for continuous action under all circumstances that is given to the duplex pump. Power pumps are often used for boiler feeding.

61. Whenever possible the boiler feeding apparatus should be in duplicate, so that the stoppage of one set will not affect the running of the plant. This end is generally secured by installing both a pump and an injector, each having a capacity sufficient for the needs of the plant.

62. Steam-driven crank-and-flywheel pumps are occasionally used, but they are open to the serious objection that they cannot always be run slow enough to suit the demand without stopping on the centers. In very large electrical installations, the electrically driven power pump is the most economical and satisfactory arrangement. Mills and factories often use the two-throw power pump having a movable crankpin, by means of which the stroke and hence the quantity of water pumped can be adjusted to suit the requirements. By this means a constant supply of feed-water equal to the demands for steam can be obtained, which is superior to the practice of pumping large quantities of water into the boilers at intervals. Boiler-feed pumps should not be required to run faster than 100 feet per minute piston speed. The velocity of water through the suction pipe should not exceed 200 feet and through the delivery should not be more than 400 feet. If the pipes are long or fitted with elbows, the velocity should be correspondingly decreased.

63. In determining the proper capacity of a pump for boiler feeding, the pump should be selected in reference to the amount of steam the boilers must supply. This is rarely only the amount used by the engine; in fact, in many industrial establishments much more steam is needed for other machinery than for the engine. Hence, it is best to always base the estimate as to the amount of water required on the maximum capacity of the boilers.

64. The maximum water consumption may be estimated in pounds per minute by one of the following rules, which hold good for average practice under natural draft. It will be observed that no rule based on the so-called "boiler horsepower" is given, for the reason that this is too variable a quantity to place any reliance on.

Rule 19.—*For plain cylindrical boilers multiply the product of the length and diameter in feet by .18.*

Rule 20.—*For tubular boilers multiply the heating surface in square feet by .06.*

Rule 21.—*Multiply the grate surface in square feet by 1.7.*

Rule 22.—*Multiply the estimated coal consumption in pounds per hour by .17.*

65. Whenever possible the feed-pumps should be located in the boiler room, so as to be directly in sight and in charge of the boiler attendant. In very large installations it is common to arrange the pumps in a separate pump house, they being then in charge of one of the assistant engineers, the boiler attendants regulating the supply to each battery by valves in the feedpipes.

GENERAL-SERVICE PUMPS.

66. **General-service pumps** are a line of pumps placed on the market by many of the pump builders to be used for any service where the water pressure does not exceed 150 pounds. They are generally of the plunger type and are built in sizes varying from those having a 4-inch to those

having a 16-inch plunger, and of a capacity varying from 100 gallons to 2,500 gallons per minute. They may be used for any service such as boiler feeding, fire, hydraulic elevator, or anywhere where the pressure to be pumped against is not greater than the limit stated.

TANK OR LIGHT-SERVICE PUMPS.

67. Tank or light-service pumps are of the same general form and interior construction as general-service pumps, except that the plungers are much larger in proportion to the steam cylinders, equalling or exceeding them in diameter. Such pumps cannot be used to feed their own boilers, but they are sometimes fitted with an attached pump for this purpose. Light-service pumps are commonly built of the same capacity as general-service pumps, but can only pump against low pressures.

FIRE PUMPS.

68. Fire pumps are most frequently of the duplex double-acting type with a ratio of area of steam cylinder to water piston of about 4 to 1. The duplex engine is chosen for this service on account of its simplicity and the peculiar adaptability of its motion to the high speed that is sometimes required in this service. A fire pump is frequently fitted up with a number of nozzles for hose connection. It should have relief valves, air and vacuum chambers of large capacity, steam and water gauges, priming pipes, and all the necessary valves.

Fire pumps, as implied by the name, are intended for use in case of fire, and are required to throw a large volume of water at high pressure.

LOW-PRESSURE STEAM PUMPS.

69. Low-pressure steam pumps are pumps intended for localities where only a low steam pressure is available, as in apartment houses, public and private buildings, etc.,

in which the pressure at which the steam heating system is worked does not exceed 5 to 10 pounds per square inch. The ratio of cylinder areas is about 9 to 1, the steam cylinder being the larger. Otherwise they are fitted up similar to pumps for general service. In some cases a hand power attachment is provided so that the pump can be worked by hand when the steam pressure is down.

PRESSURE PUMPS.

70. Pressure pumps are designed especially for use in connection with hydraulic lifts, cranes, cotton presses, testing machines, hydraulic machine tools of all kinds, and hydraulic presses, also for oil pipe lines, mining purposes, and such services as require the delivery of liquids under very heavy pressure. These pumps are invariably of the outside-packed plunger type and generally have four single-acting plungers working in the ends of the water cylinders, the latter having a central partition. The water valves are contained in small chambers capable of resisting very heavy pressures and ingeniously arranged for ready access. All materials used in the construction of the water end must be first class and suitable to the pressure used, which ranges from 750 pounds per square inch to 1,500 pounds per square inch. The water ends of these pumps are frequently made of hard, close-grained composition for medium pressures, and of steel castings for the heavier pressures.

MINE PUMPS.

71. Perhaps no other class of pump requires as much experience and skill to select as the mine pump. The reason for this is the wide variations in service, conditions of operation, head or pressure to be worked against, and the destructive nature of the water to be pumped. Nearly all the pumps at present installed are placed entirely below the surface. In former times the Cornish, or bull, pump was the favorite, but it is today abandoned for the more compact

and less expensive modern mine pump. The water end of the modern high-pressure mine pump may be described as having outside-packed plungers; strong circular valve pots independent of one another, but bolted to the working chamber, to the suction and delivery pipes, and to one another. Frequently the whole inside of the water end of the pump, from the suction nozzle to the discharge flange, is lined with wood, lead, or some other acid-resisting substance. Sometimes the entire water end is made of an acid-resisting bronze. Unless the service is light the outside-packed plunger pump is recommended for mine work; the valves should be preferably metallic valves in separate pots or chambers. Whether the pump shall be simple, compound, or triple expansion depends much on the price of fuel. In the anthracite coal regions the compound mine pump is now very common for sizes as small as 1,000,000 gallons in 24 hours, and they are invariably compounded for larger sizes, while the triple-expansion direct-acting pump is found in several of the mines.

72. Compound crank-and-flywheel high-duty pumps using the steam expansively have but recently been installed in the coal mines; in the iron and copper mines, where the cost of fuel is very high, the highest types of pumping engines have long been used.

73. When the larger types of high-duty pumps are used, the mine workings are generally so arranged that all the water runs to one large basin or sump near which a chamber of sufficient size is cut to contain the pump, which is surrounded and protected by suitable devices to maintain it in a high state of efficiency.

74. In many mines, strength and simplicity are the controlling elements in selection, for the reasons that many mines are compelled to use a large number of medium sized pumps and, for commercial reasons, use only one man whose business it is to make the rounds of the various pumps, giving each one but a few minutes' attention in a day. They generally have to stand rough usage, and the water pumped

is of such a corrosive quality that repeated renewals of parts of the water end are absolutely necessary. After heavy rains or other causes of flooding, the pumps are often required to run for days completely submerged and must pump both themselves and the mine dry. It can be readily seen that a pump for such service must be strong, simple, ready of access, and all of its parts of such construction that they can be readily taken apart or renewed.

SINKING PUMPS.

75. Sinking pumps are used in sinking or deepening mine shafts. There is little choice in their selection; generally speaking, they should be simple, strong, and capable of working in any position. The valves should be of the simplest possible construction and accessible for renewal with a minimum of labor and time. The valve motion should be simple and protected from dirt and drippings. They are regularly built single cylinder and duplex and are steam or electrically driven. With electric sinking pumps the protection of the electrical parts must be very complete.

BALLAST AND WRECKING PUMPS.

76. Ballast and **wrecking pumps** are principally confined to the marine service. The ballast pump is used on steamers having an extensive system of water ballast; also, for handling petroleum in bulk on oil-tank steamers. It is distinctively a special pump. The wrecking pump has a somewhat wider sphere. As its name implies, it is used principally by wrecking companies on the Atlantic and Pacific coasts and along the Lakes and is constructed with particular reference to reliability, portability, and general efficiency. It is well adapted to other services requiring the delivery of large volumes of water within the range of lift by suction. It has no forcing power, the water being merely delivered over the top of the pump, and it is single-acting, the water piston being fitted with valves. It is a

very light form of pump in proportion to the work it will do, is simple, durable, and not liable to derangement or breakage. It is also well adapted to drainage and irrigating purposes.

DEEP-WELL PUMPS.

77. Deep-well pumps, like sinking pumps, give little field for choice except in the pump-driving mechanism, which is as varied as the agent available to operate them, the principal agents being steam, electricity, gas, and windmills. The pump is usually a lifting pump having a bucket packed with numerous hydraulic leathers and working within the casing; it is usually given a very long stroke. These pumps do not handle gritty water successfully. Probably the best practical solution of the deep-well pump problem will be found in the air lift, which in principle and operation is quite simple.

SEWAGE PUMPS.

78. Sewage pumps are built in various types. When the lift is low, which condition is most common in sewage disposal, the centrifugal pump is the cheapest to install, but when economy and efficiency are important factors, the centrifugal pump must give place to the more expensive but more efficient reciprocating pump. Probably the largest single pumping engine ever constructed is the sewage pump for the city of Boston, which has a capacity of 70,000,000 gallons in 24 hours.

79. It will readily be seen that the selection of a type of sewage engine will depend much on the capacity of the installation and the price of fuel delivered at the station. The principal characteristics of the sewage engine are in the valves, which must be provided with very large ports to allow fairly large objects to pass through the pump without obstructing its valves. The valves are frequently made in the form of large leather-faced doors or flap valves, giving

nearly the full area of the pipe. The sewage pump does not differ in other respects from pumps for general service

---

POWER PUMPS.

80. Power pumps are among the oldest styles of pumps, and may be developed by driving any type of reciprocating pump by other means than the use of a directly attached steam, gas, or air cylinder. Power pumps are very often geared or belted and with the increasing application of electricity the electric power pump is coming into more extensive use.

81. Power pumps may be used for any service and are frequently found in municipal water works, being often driven by a turbine or a Pelton waterwheel. In large electric-lighting, heat, and power plants, the power pump is much used for boiler feeding; in this case the pumps are usually triplex, giving a steady flow of water, and are driven by electric motors, the current being furnished by the main generators. This is probably the most efficient and economical boiler feeder that has been developed.

82. The power pump is used quite extensively in the mines. An electric motor being the driver, the system admits of many various sized pumps being placed at the different sumps throughout the mines and driven by one large and economical generating unit at the surface.

83. The selection of a power pump in preference to other types depends on conditions that, to some extent, may be gathered from the above applications; the choice, however, depends much on the kind of power available to run the machine. Where water-power is available, either for gearing directly to the pump or for generating electric current to drive the pump at a remote distance, the power pump may advantageously be chosen. It should be remembered in this connection that a steam pump should be installed to take

care of the feedwater when the main engines are stopped and no current is available for driving the power pump.

84. In private houses, hotels, office and public buildings the electric-power pump is a favorite, and to avoid the noise of gears the reduction in speed is made by friction drives of various types; rawhide gearing is also used to some extent. The construction of the water end of power pumps does not differ from other pump constructions for the same service.

MUNICIPAL PUMPING ENGINES.

85. While the **municipal pumping engine** may be of any size and capacity, and while some of the pumps already discussed, as the general-service and power pump, may be, and are, frequently used, the term usually implies the highest type of pumping engine that can be constructed as regards economy and efficiency. The refinement is more exacting as the capacity of the pump increases. For small municipal pumping engines, say of 2,000,000 to 5,000,000 gallons capacity in 24 hours, the compound and triple-expansion direct-acting engine is used, the degree of expansion depending on the price of fuel and the capital available for the investment. For installations of from 5,000,000 to 20,000,000 gallons capacity, the high-duty direct-acting engine, that is, the direct-acting engine with high-duty attachment, and the crank-and-flywheel engine are rivals for the installation; while for large municipal pumping engines above 20,000,000 gallons capacity in 24 hours, the vertical triple-expansion condensing three-crank single-acting, or differential, plunger beam type may be said to have no equal. With the latter type of engine a duty of 160,000,000 foot-pounds of work per 1,000 pounds of steam used by the engine is now common. Steam pressures of 175 pounds are common, while the number of expansions are as high as 22 to 26, and every reasonable device known in the art of steam engineering is used to the end of breaking records in securing a high duty.

VACUUM PUMPS.

86. Vacuum pumps are chiefly used in connection with jet condensers and siphon condensers. A vacuum pump is in reality an air pump, it being used for pumping air out of closed vessels. There are two general types of vacuum pumps, which are **dry vacuum pumps,** or pumps that handle air only, and **wet vacuum pumps,** or pumps that handle both air and water. Vacuum pumps are also used in some manufacturing operations where a high degree of vacuum is required, being used in connection with the vacuum pans found in sugar houses, with glycerin pans, etc.

RELATIVE MERITS OF DIRECT-ACTING AND CRANK-AND-FLYWHEEL PUMPS.

87. The relative merits of the two types of machine for a particular size, other conditions being equal, are such that it is a very difficult matter to decide which type is superior. For pumping small quantities of water, say up to 700 gallons per minute, and in localities where coal is not expensive, the direct-acting pump, either simple or compound, should prove a good investment. The objection to the direct-acting pump for large sizes is its waste of steam as compared with the crank-and-flywheel pump; it has an additional objection that is sometimes argued against it, which is *short-stroking*. This defect reduces its economical performance in that it requires some steam to fill up the space due to the incomplete stroke, but since the incomplete stroke is due to too high a compression, the compressed steam must have nearly filled the space before fresh steam was admitted, so that the loss is not so very great after all. Short-stroking reduces the capacity of the machine somewhat. In the common types of direct-acting pumps, the steam is not worked expansively; in compound and triple-expansion pumps, some degree of expansion is obtained, usually a little more than the ratio of low-pressure cylinder to high-pressure cylinder. By making the reciprocating parts

heavy and running the pump at some fixed minimum speed, an early cut-off can be effected in the high-pressure cylinder, the balance of the stroke being completed by the inertia of the reciprocating parts; in this way an increased degree of expansion is possible.

Another method of securing a considerable degree of expansion in the direct-acting pump is by means of the high-duty attachment. With the same degree of safety the speed of the direct-acting pump is very much less than is possible with the crank-and-flywheel pump. The direct-acting pump in which any attempt is made at economy will occupy quite as much space as the crank-and-flywheel pump of the same capacity, but the direct-acting pump is lower in first cost.

88. Probably the most objectionable feature of the crank-and-flywheel pump, which is an inherent one, is that the velocity of discharge varies throughout the stroke. This is due to the fact that while the flywheel rotates at a uniform speed, the pistons and plungers move with a variable speed, varying from zero at the beginning of the stroke to the maximum speed near mid-stroke and then decreasing to zero at the end of the stroke. This variation in velocity produces shocks, and hence requires the water end of a flywheel pump to be of heavier construction than a similar end for a direct-acting pump. The valve area of a flywheel pump requires to be considerably larger than for a direct-acting pump, not only because of its capacity for higher speeds, but also because the velocity of the plunger, when the connecting-rod is at right angles to the crank arm, is somewhat in excess of 1.57 times the mean velocity of the plunger. In addition to the greater valve area and strength required in flywheel pumps, it is necessary to use some means to reduce the shocks to the mechanism and parts of the pump. This is accomplished by providing large air chambers, preferably one over each deck for high pressures; for very high pressures and long columns of water, an alleviator is necessary.

89. The main advantage of the crank-and-flywheel pump is its economy, which, in turn, is due to the fact that the

steam may be expanded to any permissible degree; it also readily admits of all the refinements known of securing high-duty performance, and with a proper arrangement of details, it can be made quite as safe as ordinary machines. For extreme high duties the crank-and-flywheel pump is always chosen, and to reduce the shocks due to a variable discharge a favorite type is the three-crank machine. The combined delivery from three plungers is tolerably uniform and the arrangement readily lends itself to the extremely economical triple-expansion condensing engine.

90. The crank-and-flywheel engine is more expensive than the direct-acting machine, and when high degrees of expansion are used occupies considerably more room. It is generally more complicated, but is more accessible, except in such cases as where an effort is made to minimize space, when by making the engine back-acting it is liable to become quite inaccessible.

91. The piston speed of direct-acting pumps rarely exceeds 100 feet per minute, while the piston speed of crank-and-flywheel pumps is commonly 300 feet and sometimes 400 feet. With pumps of the controlled-valve type, piston speeds of 560 feet are reached. This difference in the piston speed of the direct-acting and crank-and-flywheel pumps shows that they must be compared on the basis of water delivered rather than on the relative size of similar parts.

92. Even for very small sizes, the crank pump is sure in its action and is not liable to get out of order; this cannot be claimed for some of the single-cylinder direct-acting pumps having steam-thrown valves. The crank pump is limited as to its slowest speed, however, since the speed must be sufficient to store energy enough in the flywheel to carry the crank over the dead centers. This objection can be overcome to a great extent by using the by-pass, which allows part of the water to be returned to the suction, thus decreasing the work on the pump.

A SERIES OF QUESTIONS AND EXAMPLES

Relating to the Subjects Treated of in This Volume

It will be noticed that the Examination Questions that follow have been divided into sections, which have been given the same numbers as the Instruction Papers to which they refer. No attempt should be made to answer any of the questions or to solve any of the examples until that portion of the text having the same section number as the section in which the questions or examples occur has been carefully studied.

THE STEAM ENGINE.

EXAMINATION QUESTIONS.

(1) Define the terms head end and crank end of a steam-engine cylinder.

(2) What is meant by the stroke of an engine?

(3) What are the stationary parts of an engine?

(4) Define *valve gear.*

(5) Explain the difference, if there is any, between piston clearance and clearance volume.

(6) What is understood by the throw of an eccentric?

(7) Explain the difference between outside lap and inside lap of a **D** slide valve.

(8) Is it possible to cut off when a **D** slide valve operated by an eccentric has no outside lap and no lead?

(9) What is meant by angle of advance?

(10) Suppose you had a plain slide-valve engine and you wished to make the cut-off earlier, what would you do? The port opening is to be the same as before.

(11) What is the effect of increasing the inside lap of a **D** slide valve?

(12) Define *lead.*

(13) With an ordinary slide valve and an engine running under, is the eccentric set behind or ahead of the crank?

(14) How can the valve be given a travel greater than the throw of the eccentric?

(15) If a reversing rocker is used with an ordinary slide valve, will the eccentric occupy the same position as with a direct rocker?

(16) How does the angularity of the connecting-rod affect compression?

(17) What is the object of the passage cored in an Allen valve?

(18) State in your own words how to set the valve of a plain slide-valve engine.

(19) A $14'' \times 28''$ engine has a clearance volume of 247 cubic inches. Express the clearance in per cent.

Ans. 5.73 per cent.

(20) In a $36'' \times 60''$ engine the steam is cut off when the piston has moved over 21 inches of its stroke. The clearance being 2 per cent., find the real cut-off.

Ans. 36.27 per cent.

(21) What is the ratio of expansion of the engine given in question 20? Ans. 2.76.

(22) Suppose that the outside lap of a D slide valve is decreased, but that the valve travel and angle of advance remain the same as before. Investigate the effect of this with the aid of the Bilgram valve diagram and state your conclusions and explain how they were reached.

(23) A $44'' \times 80''$ engine is to run at 75 revolutions per minute. What actual diameter of steam and exhaust pipe should be used?

$$\text{Ans.} \begin{cases} \text{Steam pipe} = 18 \text{ in.} \\ \text{Exhaust pipe} = 22 \text{ in.} \end{cases}$$

(24) What should be the area of the steam port for the engine given in question 23 if the steam port is short?

Ans. 202.7 sq. in.

THE INDICATOR

EXAMINATION QUESTIONS.

(1) What is meant by the scale of an indicator spring?

(2) About what scale of spring is usually selected for a given boiler pressure?

(3) What is a reducing motion?

(4) What is the principal objection to the lazy-tongs and pantograph reducing motions?

(5) In Fig. 9, find the length of the arm *U V* so that the diagram may be 3 inches long, the stroke of the engine being 24 inches, and the length of the arm *U W* 40 inches.

Ans. 5 in.

(6) What precautions should be taken before attaching an indicator to an engine?

(7) What is the vacuum line and how is its position located?

(8) What are the distinguishing characteristics of indicator diagrams taken from Corliss engines as compared with diagrams taken from high-speed engines?

(9) (*a*) In a plain slide-valve engine, how would you remedy too early admission? (*b*) What effect would this remedy have on the other events of the stroke?

(10) If one end of a cylinder with a slide valve is found to be doing more work than the other, how can the fault be remedied?

(11) (*a*) How is the amount of compression influenced by the speed of the engine? (*b*) What should be the amount of compression for high-, low-, and medium-speed engines?

(12) What may be inferred (*a*) when the steam line falls abruptly? (*b*) when the back-pressure line is much above the atmospheric line? (*c*) when the actual expansion line rises above the theoretical expansion line?

(13) (*a*) What faults in steam distribution are shown by the diagam, Fig. I, which is taken from a plain slide-valve

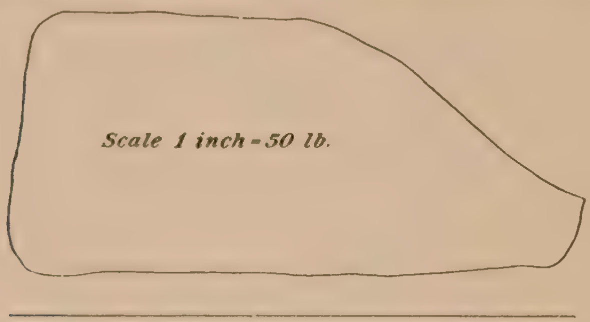


FIG. I.

engine? (*b*) How may they be partly remedied?

(14) Criticize the indicator diagrams shown in Fig. II.

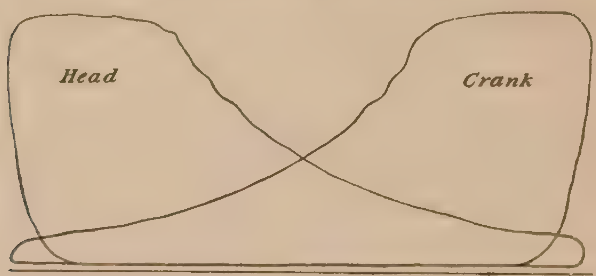


FIG. II.

(15) Why does the actual expansion line generally rise above the theoretical expansion line near the end of the stroke?

(16) What is the most general method of determining the point of cut-off on a diagram taken from a high-speed engine?

(17) (*a*) To what are wavy lines on a diagram generally due? (*b*) Expansion lines that drop by a series of steps?

(18) Criticize the diagrams shown in Fig. III.

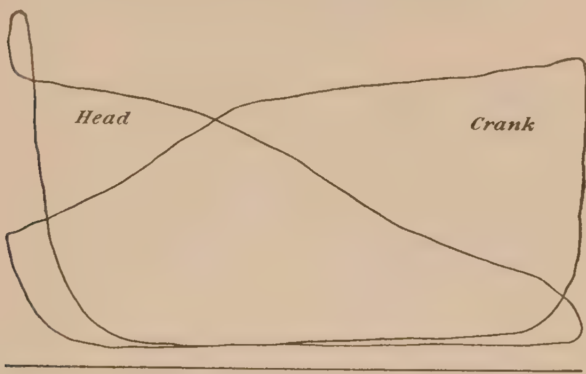


FIG. III.

(19) What is the cause of the difference in the shape of the loop above the steam line in Figs. 20 and 22?

(20) If the actual expansion line follows the theoretical expansion line closely, is that a positive indication that the valves and piston do not leak?

FIG. IV.

(21) Criticize the diagrams shown in Fig. IV.

ENGINE TESTING.

EXAMINATION QUESTIONS.

(1) Define (*a*) adiabatic expansion; (*b*) isothermal expansion.

(2) What relation exists between the amount of work done in compressing a gas isothermally or adiabatically and the amount of work done by the gas when expanding under similar conditions?

(3) Why is not the relation of volume and pressure of steam when it expands as simple as in the case of a perfect gas?

(4) If the net pressure on the piston of an engine is 45.6 pounds per square inch and the volume swept through by the piston at each stroke is 6.3 cubic feet, (*a*) how much work is done at each stroke? (*b*) The engine makes 76 strokes per minute; what horsepower does it develop?

Ans. { 41,368.32 ft.-lb.
95.27 H. P.

(5) What is (*a*) the mean ordinate of an indicator diagram and (*b*) how is it found?

(6) A diagram like that shown in Fig. 7 is 3 inches long and has an area of 6.75 square inches; the vertical scale of pressure is 50 pounds per inch; the cylinder from which the diagram was made has an area of 1 square foot and a length of 2 feet. (*a*) Find the horizontal scale of volumes and (*b*) the work per stroke of piston.

Ans. { (*a*) $\frac{2}{3}$ cu. ft. per in.
(*b*) 32,400 ft.-lb.

(7) How can the net horsepower of an engine be approximately obtained without the use of a dynamometer?

(8) The I. H. P. of an engine running under full load is 176.8. When running light the I. H. P. is 25.6. What is the efficiency of the engine? Ans. 85.5 per cent.

(9) The area of a diagram is 2.75 square inches and the length is 3.15 inches. A 50-pound spring was used. Find the M. E. P. Ans. 43.65 lb. per sq. in.

(10) In finding the area of a diagram with a planimeter, how may the accuracy of the work be easily checked?

(11) If the M. E. P. of a diagram with loops is to be found by the use of ordinates, how can the mean ordinate be found?

(12) Find the approximate M. E. P. of a non-condensing engine cutting off at $\frac{5}{8}$ stroke and making 300 revolutions per minute. The boiler pressure is 75 pounds gauge.

Ans. 59.54 lb. per sq. in.

(13) (*a*) What is meant by piston speed? (*b*) An engine with a 24-inch stroke runs at a speed of 180 revolutions per minute. What is the piston speed? Ans. 720 ft. per min.

(14) What is an engine constant?

(15) (*a*) What is the engine constant for a uniform speed of rotation of an $18'' \times 24''$ engine running at a speed of 185 R. P. M.? (*b*) What is the I. H. P. of the engine when the average M. E. P. for a pair of indicator diagrams is 53.8 pounds per square inch?

Ans. $\begin{cases} (a) & 5.706. \\ (b) & 306.98. \end{cases}$

(16) (*a*) Find the engine constant for a uniform scale, number of ordinates and piston speed of a $24'' \times 36''$ engine running at 150 revolutions per minute when 20 ordinates are used and the scale of the spring is 60. (*b*) What is the I. H. P. of the engine when one-half the sum of the lengths of the ordinates is 16 inches?

Ans. $\begin{cases} (a) & 37.013. \\ (b) & 592.2. \end{cases}$

(17) What is meant by the brake horsepower of an engine?

(18) In Fig. 17 the distance from the center of the shaft to the point of support of the brake arm on the scale is 4 feet. When the brake is not in operation the scale balances at 14.5 pounds. What horsepower is developed by the engine when it is running at 225 revolutions per minute and the scale balances at 274.5 pounds? Ans. 44.55 H. P.

(19) In Fig. 18 the diameter of the pulley is 47 inches and the diameter of the rope is 1 inch. When the engine is running at 350 revolutions per minute, the weight W is 241 pounds and the spring balance A indicates 10 pounds. What horsepower is developed? Ans. 30.78 H. P.

(20) Why is the steam consumption, as calculated from an indicator diagram, always less than the actual steam consumption?

(21) How can an idea of the amount of cylinder condensation be obtained?

(22) The following measurements were taken from a diagram like Fig. 19: $a\,m = .70$ inch, $O\,m = 3.41$ inches, $b\,n = .62$ inch, $O\,n = .36$ inch, and $e\,h = 3.25$ inches. The diagram was taken from an engine having a $16'' \times 20''$ cylinder and running at 160 revolutions per minute. The area of the diagram is 2.41 square inches and the scale of the spring is 45 pounds. Find the steam consumption per I. H. P. per hour. Ans. 30.34 lb.

(23) The following measurements were obtained from a diagram (see Fig. 20) taken from a $24'' \times 36''$ engine: $a\,m = .71$ inch, $l = 2.93$ inches, $L = 3.36$ inches, M. E. P. $= 37.5$ pounds, spring 50. What is the steam consumption per I. H. P. per hour? Ans. 27.79 lb.

GOVERNORS

EXAMINATION QUESTIONS

(1) Explain the necessity of a governor for steam engines and steam turbines.

(2) How does the flywheel of an engine assist the governor?

(3) What is the purpose of an emergency governor?

(4) State what is the fundamental difference between a speed governor and a pressure governor.

(5) What is the advantage of the weighted pendulum governor?

(6) In the pendulum governor, what balances the centrifugal force?

(7) Name the two general types of shaft governors.

(8) Does centrifugal force enter into the operation of inertia shaft governors?

(9) What is a governor case?

(10) What is a dashpot used for in connection with engine governors?

(11) Give an example of an engine governor that cnanges only the angle of advance of the eccentric, but not the throw.

(12) In the Buckeye engine governor, what is the object of the auxiliary springs?

(13) Why are some engine governors equipped with an adjustable weight?

(14) Will changing the size of the governor pulley make any difference in the speed of the engine?

(15) Is it allowable to make moderate changes of engine speed by changing the spring tension in shaft governors?

(16) If a shaft governor is slow in acting, how may it be improved?

(17) What is the idea in making the weights of shaft governors hollow?

(18) Before attempting to adjust a shaft governor that does not work satisfactorily, what should be done?

(19) In the Mason pressure regulator does steam or water actuate the diaphragm?

(20) What condition is necessary for the satisfactory working of a pump governor that is to maintain a uniform water level in a steam boiler?

VALVE GEARS.

(PART 1.)

EXAMINATION QUESTIONS.

(1) What are the leading types of valves?

(2) What is the object of using expansion valves?

(3) Mention two disadvantages of the Gonzenbach expansion valve.

(4) In what position relative to the crank are the eccentrics set for the Meyer cut-off valve?

(5) If the auxiliary valve of a riding cut-off valve is non-adjustable, how may its eccentric be set?

(6) Mention some of the disadvantages of a plain D slide valve when used as an automatic cut-off engine.

(7) What advantages has the pressure-plate valve over the piston valve?

(8) What precaution is necessary in starting an engine with piston valves or pressure-plate valves?

(9) If live steam should leak past the packing strips of an Allen balanced valve, would the balancing of the valve be affected?

(10) What is the function of the dashpot on a Corliss engine?

(11) The motion imparted to the valves of a Corliss valve gear by a given angular motion of the wristplate varies through wide limits. How is this variation made to improve the action of the valves?

(12) If the load on a Corliss engine becomes so heavy that the admission valve is not released, how will the valve work?

(13) Explain the necessity of lead and lap on Corliss valves.

(14) Why are Corliss valve gears sometimes made with two eccentrics?

(15) If two eccentrics are used on a Corliss valve gear, what is the effect on the opening of the steam valves if the steam eccentric is set too near the crank position?

(16) Mention some advantages of the high-speed Corliss valve gear as compared with the ordinary Corliss gear.

(17) Suppose the wristplate of a Corliss gear overtravels at one end of its motion; how can this fault be remedied? The rocker-arm is supposed to be properly set.

(18) What effect on the events of stroke has increasing the distance between the valve plates of a Meyer cut-off valve?

(19) What do you understand by "overtravel" of a valve?

(20) Why are multiple-ported valves used?

(21) On an engine using the Meyer cut-off valve, admission occurs too early on the crank end, but is correct on the head end. What must be done in order that admission may occur at the correct time at both ends of the cylinder?

VALVE GEARS.

(PART 2.)

EXAMINATION QUESTIONS.

(1) When the hook, or gab, gear is used, is the cut-off adjustable?

(2) What is the principal difference between the Stephenson and Gooch link motions?

(3) In determining whether eccentric rods are open or crossed, what should be the position of the eccentrics with reference to the link?

(4) If the link of a Stephenson link motion using open rods is in mid-gear, will steam be admitted to the cylinder during any part of the stroke?

(5) If the block of a Gooch link motion is shifted towards the mid-gear position, how is the lead affected?

(6) In setting a valve with a link motion, how is the lead equalized for the second full-gear position?

(7) Why is a valve driven by a Stephenson link motion sometimes set when the link is in the ordinary running position instead of the full-gear position?

(8) Why are radial valve gears but little used on automatic cut-off engines?

(9) Can the lead of a valve with a Joy or See-Marshall valve gear be readily changed?

(10) Give the advantages and disadvantages of the single- and double-seat poppet valves.

(11) (*a*) Why can an almost perfect steam distribution be obtained with cam gears? (*b*) Why can they not be used at high speeds?

(12) What are the advantages of gridiron valves?

(13) (*a*) Is it generally possible to set the valves of an automatic cut-off engine so that cut-off will be equal for both ends of the cylinder for different loads? (*b*) How are some gears designed so as to give equal leads and equal cut-offs?

(14) In vertical engines, why is it well to give the valve more lead for the lower port than is given for the upper one?

(15) Describe the method of locating the marks for determining the position of a valve not adjustable along the stem.

(16) How would you set a plain slide valve if a See-Marshall valve gear is used?

(17) How does the Joy valve gear differ from the Stephenson link motion with open rods in respect to the effect of early cut-off on the lead and compression?

(18) Are both ports opened the same distance with a See-Marshall valve gear?

(19) How will the lead be affected when the link is shifted towards mid-gear (*a*) if open rods are used? (*b*) if crossed rods are used?

(20) With the McIntosh & Seymour valve gear, what is the effect of the togglejoint on the motion of the valves?

CONDENSERS.

EXAMINATION QUESTIONS.

(1) Why is it that a perfect vacuum cannot be formed in a condenser?

(2) What is the function of the air pump attached to a condenser?

(3) State the principle of condensing the steam (*a*) in a jet condenser and (*b*) in a surface condenser.

(4) If a condenser becomes flooded, what is liable to happen to the engine?

(5) When does a condenser get hot?

(6) Briefly explain the theory of the condenser.

(7) If an engine exhausts into a condenser, what is the only sure way of telling if the steam valves and pistons are tight?

(8) Mention two ways in which the amount of steam generally used in large plants to run the independent air and circulating pumps may be reduced.

(9) In a siphon condenser, how is the vacuum formed?

(10) What is the object of contracting the pipe of a siphon condenser into a neck, or throat?

(11) (*a*) A siphon condenser, like the Baragwanath, is placed 34 feet above the hotwell. Is it necessary for the pump to force the injection water to this height? (*b*) Why?

(12) (*a*) If the supply of injection water is impure, what type of condenser should be used? (*b*) Why?

(13) (*a*) How many pumps are required to operate a surface condenser? (*b*) What is the function of each?

(14) What are some of the advantages and disadvantages of the surface condenser compared with the jet condenser?

(15) What is the use of the snifting valve attached to a surface condenser?

(16) Where a surface condenser is used, why is it objectionable to take feedwater from the circulating side of the condenser to supply that lost by leakage, blowing-off, etc.?

(17) (*a*) What is the cause of loss of efficiency in surface condensers? (*b*) How may the efficiency be restored? (*c*) How may the loss of efficiency be prevented?

(18) (*a*) What is a good composition for condenser tubes? (*b*) How should the surface of the tubes be protected?

(19) (*a*) How are condenser tubes generally fastened in the tube-sheets? (*b*) Why are they so fastened?

(20) How may split condenser tubes and leaky tube packings be detected?

(21) How will air leaks manifest themselves in a condenser?

(22) What is the object of tinning condenser tubes both inside and outside?

(23) On what principle do all devices for cooling condensing water operate?

(24) When water is freely exposed to the air, on what factors does the amount of evaporation depend?

(25) Explain how the evaporation of a portion of the condensing water cools the remaining portion.

(26) Briefly describe the principle of the cooling tower.

(27) (*a*) In what three ways does the warm water falling through a cooling tower lose its heat? (*b*) Which has the greatest cooling effect?

(28) Briefly describe the Linde system of cooling condensing water.

(29) If the valves of a circulating pump which is driven from the main engine slam on account of too little water being pumped, how may the slamming be stopped?

(30) How is it usual to provide against the breaking of a cylinder head should the air pump of a jet condenser suddenly refuse to work and allow the water to back up towards the cylinder?

(31) On what does the amount of water required to condense a pound of steam depend?

(32) The temperature of the water entering a surface condenser is 55° and on leaving its temperature is 105°. The pressure of the steam at release is 5 pounds absolute and the temperature of the condensed steam as it enters the air pump is 135°. How many pounds of condensing water are required per pound of steam? Ans. 20.57 lb., nearly.

(33) If the vacuum in a jet condenser is less than it should be and the hotwell temperature is higher, what is probably the trouble?

(34) The exhaust enters a jet condenser at a pressure of 3 pounds absolute. The temperature of the condensing water is 65° and the temperature of the mixture as it enters the pump is 130°. How much condensing water is used per pound of steam? Ans. 15.8 lb.

(35) In practice about how much vacuum can be obtained?

COMPOUND ENGINES.

EXAMINATION QUESTIONS.

(1) Mention some of the mechanical advantages that the compound engine has over the single-cylinder engine of equal power.

(2) Why cannot a high ratio of expansion be economically used in a single cylinder?

(3) State briefly why the division of the temperature range between several cylinders tends to reduce cylinder condensation.

(4) Is it probable that the steam condensed in the high-pressure cylinder does any work in the low-pressure cylinder?

(5) (*a*) Which requires the heavier flywheel for equal steadiness, a tandem compound engine or a cross-compound engine? (*b*) Why?

(6) Briefly distinguish between the Woolf compound type and the receiver compound type of steam engines.

(7) Explain why a receiver is necessary for a cross-compound engine having cranks set 90° apart.

(8) To what do the terms compound, triple-expansion, and quadruple-expansion engine refer?

(9) Mention some of the advantages claimed for the so-called triangular connecting-rod patented by John Musgrave & Sons.

(10) What do you understand by the term "drop" when applied to the pressure in a receiver?

(11) What effect has the drop in a receiver on the quality of the steam that enters it?

(12) (*a*) How does a change in the high-pressure cut-off affect the receiver pressure? (*b*) Give reasons for your answer.

(13) How may the drop in a receiver be regulated?

(14) If the receiver pressure is raised by changing the low-pressure cut-off, what effect will it have on the relative amount of work done in the two cylinders?

(15) What objection is there to governing a compound engine by changing only the high-pressure cut-off?

(16) Why should a pop safety valve be fitted to the receiver?

(17) (*a*) In a cross-compound engine having duplicate piston rods, connecting-rods, crankpins, and crossheads, the power developed by the two cylinders is the same. Will the stresses in the duplicate parts necessarily be the same? (*b*) Why?

(18) In practice how is it probably best to distribute the work between the cylinders?

(19) (*a*) What is the object of the steam jacket? (*b*) Will anything be gained by using a steam jacket when superheated steam is used? (*c*) Why?

(20) In a horizontal compound engine should the high-pressure or the low-pressure valves have the most lead?

(21) (*a*) What is a reheater? (*b*) What is its purpose?

(22) In using a reheater the temperature of the exhaust is found to be greater than that corresponding to its pressure. What does this indicate?

(23) If a reheater is constructed similar to a tubular boiler, why is it a good plan to take the live steam for the reheater from a connection placed between the engine and throttle?

(24) (*a*) To what is the ratio of expansion of a multiple-expansion engine equal? (*b*) Does the volume of the intermediate cylinders affect this ratio?

(25) The volume of the low-pressure cylinder of a compound engine is 12,460 cubic inches. The volume swept through by the high-pressure piston, including clearance, is 3,115 cubic inches. The ratio of expansion of the high-pressure cylinder is 3. What is the total ratio of expansion?

Ans. 12.

(26) In what two ways may the ratio of expansion of a multiple-expansion engine be expressed?

(27) The volume of the high-pressure cylinder of a compound engine up to the point of cut-off is 800 cubic inches; the volume of the low-pressure cylinder up to release is 8,800 cubic inches. What is the total ratio of expansion?

Ans. 11.

(28) Estimate the probable horsepower of a triple-expansion engine having cylinder diameters of 15, 26, and 39 inches. The common stroke is 24 inches and the speed is 180 revolutions per minute. The boiler pressure is 180 pounds, the engine is condensing and is fitted with slide valves.

Ans. 521 H. P.

ENGINE MANAGEMENT.

(PART 1.)

EXAMINATION QUESTIONS.

(1) Explain how water hammer is caused by suddenly opening a stop-valve or throttle valve and allowing steam to enter a cold steam pipe or cylinder.

(2) What precaution should be taken with regard to the throttle valve before steam is allowed to enter the main steam pipe?

(3) How should an engine be cared for after it has been stopped?

(4) Describe briefly the manner of warming up the steam pipes and engine preparatory to starting up.

(5) If the boilers are fired up just previous to starting the engine, how may the engine and steam pipe be warmed up without the use of steam?

(6) From what position of the crank is it easiest to start a horizontal engine?

(7) In starting a slide-valve non-condensing engine, why should the throttle be opened quickly?

(8) If a cracking noise is heard in the cylinder soon after starting, what is probably the trouble?

(9) Mention one advantage of allowing an engine to cool down with the cylinder drain cocks closed.

(10) How may the cylinder of an engine that is fitted with a reversing gear be warmed up?

(11) Why is it advisable to have the air and circulating pumps of a condenser driven independent of each other?

(12) Describe briefly the manner of starting a slide-valve condensing engine that is fitted with a surface condenser.

(13) (*a*) Why must the injection valve of a jet condenser be opened at the same moment that the engine is started? (*b*) If the condenser gets "hot," how may it be started?

(14) (*a*) What is a snifting valve and (*b*) what is its purpose?

(15) Briefly describe the operation of starting a Corliss engine.

(16) What is the principal difference in the manner of stopping a simple Corliss engine and a simple slide-valve engine?

(17) In general, how may the low-pressure cylinder of a compound engine be warmed up?

(18) How will too low a pressure in the receiver of a compound engine affect its starting?

(19) (*a*) For what class of compound condensing engines is the use of an independent vacuum engine particularly advantageous? (*b*) Why?

(20) If a reversible engine is fitted with an adjustable cut-off gear, how should the gear be set as soon as the engine is stopped?

(21) What relation should the center lines of the connecting-rod brasses bear to each other and to the center line of the connecting-rod?

(22) Briefly describe the method of stretching a line coincident with the center line of a cylinder, when an engine is being lined up.

(23) Suppose that it is desired to level up one of the lines used in lining up an engine; how may this be done with the aid of a plumb-line?

(24) (*a*) In lining up new shaft-bearing brasses for a horizontal engine, how is the height of the center line of the shaft usually left with regard to the center line of the cylinder? (*b*) Why is this done?

(25) Describe the method of fitting the shaft to the brasses after the latter have been lined up.

(26) In lining up an engine, suppose the crank-shaft to be in position and that it is desired to further test the shaft for being at right angles to the center line of the cylinder. How may this be done by means of the line stretched through the center of the cylinder?

(27) Describe one method of testing the crankpin for parallelism with the crank-shaft.

(28) How may it be ascertained whether or not the center lines of the connecting-rod brasses are in the same plane?

(29) What is probably the most frequent cause of pounding in engines?

(30) How may the necessity of stripping brass-bound boxes be provided against?

(31) Why is a loose piston nut very liable to cause a breakdown?

(32) Explain how too little compression may cause an engine to pound.

(33) (*a*) Will too late a release cause an engine to pound? (*b*) Give a reason for your answer.

(34) In case it is desirable not to allow sufficient water to enter the circulating pump to stop its pounding, how may the pounding be stopped?

ENGINE MANAGEMENT.

(PART 2.)

EXAMINATION QUESTIONS.

(1) What are the arguments for and against (*a*) solid bearings? (*b*) adjustable bearings?

(2) If a bearing shows an inclination to heat, how should it be treated?

(3) What is the objection to pouring cold water on hot bearings?

(4) If a bearing becomes exceedingly hot and the engine cannot be stopped long enough to allow the bearing to cool, what should be done in order to continue running the engine?

(5) What is meant by "wearing down a bearing"?

(6) If the brasses of a large journal are removed, why is the bearing very apt to heat up after the brasses have been replaced?

(7) (*a*) May loose brasses cause a bearing to heat? (*b*) How?

(8) Describe two methods of setting up bearings.

(9) What is the remedy for warped brasses?

(10) What is usually the cause of the chronic heating of bearings?

(11) (*a*) Explain why brasses that have been quickly and excessively heated are liable to pinch the journal near their edges. (*b*) How may this be prevented?

(12) How should the heating of bearings due to gritty or dirty oil be guarded against?

(13) (*a*) What one property in particular should oil for large bearings possess? (*b*) Why?

(14) In large bearings how should the bearing surfaces of the brasses be finished in order to aid the oil to penetrate between the journal and brasses?

(15) On what does the pressure that a bearing will sustain per square inch of rubbing surface without heating depend?

(16) Explain why an overloaded engine may cause the bearings to heat.

(17) Why is a little side play in a journal a good thing?

(18) On what does the value of a lubricant depend?

(19) Mention some of the desirable features of a good lubricating oil.

(20) (*a*) Can the lubricative qualities of an oil always be judged by its specific gravity or its viscosity? (*b*) Why?

(21) (*a*) What is the best animal oil for lubricating machinery? (*b*) What is probably the best vegetable oil?

(22) (*a*) What are the sources of mineral oils, and (*b*) how are they graded?

(23) Why should a compounded oil having a mineral oil base not be used as a cylinder lubricant?

(24) How are boiled, or cup, greases made?

(25) How does a lubricant prevent the rubbing surfaces becoming hot?

(26) Why are mineral oils especially adapted to lubricating pistons and slide valves working under a high steam pressure?

(27) How does the temperature of a bearing affect the lubricating power of an oil?

(28) How may the comparative viscosity of greases be approximately judged?

(29) What is meant by the flashing point of an oil?

(30) If the purity of a mineral oil is doubtful, how may it be determined if animal or vegetable oils have been mixed with it?

(31) What property of an oil is increased by adulterating it with paraffin, waxes, gums, etc.?

(32) If a mineral oil is darkened in color after it has been heated to 300° F. for a few minutes, what adulterant is probably contained in the oil?

(33) (*a*) Explain the principle of the water-displacement lubricator. (*b*) What are the objections to the use of this lubricator?

(34) (*a*) Should a double-connection lubricator ever have one connection attached to the steam pipe between the throttle and boiler and the other between the throttle and engine? (*b*) Why?

(35) (*a*) Into what three classes may steam lubricators be divided? (*b*) On what does the operation of the hydrostatic lubricator depend?

(36) What is the principal difference between double- and single-connection lubricators?

ENGINE INSTALLATION.

EXAMINATION QUESTIONS.

(1) (*a*) What are the two principal features that dictate the use of vertical engines? (*b*) Why are these the controlling features?

(2) Mention some of the most important advantages the horizontal engine has over the vertical engine.

(3) (*a*) Can the automatic cut-off engine be made to give as good steam economy as the releasing-gear engine? (*b*) If so, what is the type of the automatic cut-off engine?

(4) Are there any throttling engines built that give a fairly economical steam consumption?

(5) What two leading factors determine the use of compound engines?

(6) In deciding between the use of a simple or compound engine, what elements, besides first cost and economy of fuel, should be considered?

(7) What is the principal disadvantage of tandem compound engines?

(8) Explain why the mechanical efficiency of a cross-compound engine is greater than the efficiency of a tandem compound engine of equal power.

(9) What is the average proportion existing between the volume of the cylinders of tandem and cross-compound engine and the volume and reheating surface of the reheating receiver?

(10) Why do tandem compound condensing engines generally run smoother than cross-compound condensing engines of equal power?

(11) Mention the principal advantages and disadvantages of high-speed engines.

(12) (*a*) What two forms of valves are principally used on high-speed engines? (*b*) What are their relative advantages and disadvantages?

(13) How do high- and low-speed engines compare in regard to closeness of regulation?

(14) Explain one manner in which the fast rotative speed of high-speed engines is conducive to economy in steam consumption.

(15) Mention some of the advantages of high-speed engines for direct-connected work.

(16) In the slow-speed engine, what means are taken to secure extreme economy of steam and high mechanical efficiency?

(17) (*a*) What is the principal advantage of high-speed engines for direct-connected electrical work? (*b*) Why are they not used for large direct-connected units?

(18) If an engine of 150 horsepower is to be direct-connected to an electric generator, what type of engine should be selected to give a maximum efficiency? The steam pressure is to be 120 pounds; there is plenty of water available, but the cost of fuel is high.

(19) Why is the compound condensing engine better adapted to continuous running with varying loads than the compound non-condensing engine?

(20) Why is a simple non-condensing engine usually selected when the power required is continuous and uniform, but is liable to be increased owing to a growth in the business?

(21) What influence has the cost of fuel on the type of engine to be selected for any particular service?

(22) If superheated steam is used, will jacketing the high-pressure cylinder increase the economy of the engine ?

(23) What is the general principle of the cooling tower ?

(24) Mention some of the advantages gained by installing a number of small engines instead of one large one in a manufacturing plant.

(25) How are the vibrations of engines used on the upper floors of buildings sometimes absorbed ?

(26) (*a*) Of what materials are engine foundations usually made ? (*b*) What kind of mortar should be used ?

(27) (*a*) What is the objection to building an engine foundation directly on solid rock ? (*b*) How is this objection usually overcome ?

(28) How are engine foundations usually constructed where it is necessary to go to a great depth in order to find a sufficiently hard bottom to support the load ?

(29) What is the object of setting the capstone for the outboard bearing of an engine lower than the actual figures called for ?

(30) How are the foundation bolts for an engine located and held in position while the masonry is being built ?

(31) After an engine is in place on its foundation, how is the irregular space between the bed and foundation usually filled ?

PUMPS.

(PART 1.)

EXAMINATION QUESTIONS.

(1) Why cannot a perfect vacuum be formed in the suction chamber of a pump?

(2) Why can a suction pump at the bottom of a deep mine lift water higher than a pump at the surface?

(3) Why is it difficult to pump hot water?

(4) What do you understand by a direct-acting steam pump?

(5) What governs the speed of a direct-acting pump?

(6) What great objection to the single direct-acting steam pump does the duplex direct-acting steam pump overcome?

(7) What is the peculiarity in the manner of operating the steam valves of a duplex direct-acting steam pump?

(8) (*a*) Why is not the ordinary direct-acting pump an economical machine? (*b*) How may its economy be increased?

(9) In a Worthington slide-valve duplex pump, how is steam retained in the cylinder near the end of the stroke to form a cushion for the piston?

(10) How do the dash relief valves of a Worthington steam pump control the length of the stroke?

(11) (*a*) What is the purpose of the so-called cross-exhaust connection in compound direct-acting steam pumps? (*b*) Explain its action.

(12) What is the purpose of the high-duty attachment to direct-acting steam pumps?

(13) (*a*) Briefly explain the principle of the high-duty attachment. (*b*) In what respect is the action of the high-duty attachment better than that of a flywheel?

(14) In a pump of the Leavitt design, what is gained by making the stroke of the plunger shorter than the stroke of the engine?

(15) Describe the Quimby screw pump in your own words.

(16) (*a*) On what does the action of a centrifugal pump depend? (*b*) To what classes of work are they particularly adapted?

(17) How are the cranks of duplex double-acting power pumps set so as to give a steady flow?

(18) If the supply of power is steady, why is the belt-driven pump the most economical pump to use?

(19) Why is the plunger pump the most common type of pump used in mines?

(20) What is a pit pump?

(21) What enables the steam to be used expansively in a Cornish pumping engine?

(22) What advantages does the Bull pumping engine possess over the Cornish pump?

(23) Mention some of the objections to lift pumps when used for mining purposes.

(24) What terms are usually applied to the suction pipe and the delivery pipe of pit pumps?

(25) What is meant by a sinking pump?

(26) What are the advantages of the pulsometer?

(27) Mention some of the advantages claimed for the Pohlé air lift.

(28) Briefly describe the principle and action of a differential pump.

(29) What is the particular feature of the Riedler express pump that allows the pump to be run at such high speeds?

PUMPS.

(PART 2.)

EXAMINATION QUESTIONS.

(1) (*a*) How are the plungers of pumps used for moderate pressures usually packed? (*b*) How are they packed when used for heavy pressures?

(2) Mention two important disadvantages of the inside-packed plunger pump.

(3) What is meant by the valve deck of a pump?

(4) What is the object of curving the wings of wing valves?

(5) Why are air chambers used on the discharge pipes of pumps?

(6) (*a*) Briefly describe an alleviator and (*b*) state why they are used instead of air chambers.

(7) Why are vacuum chambers sometimes required on the suction pipe of pumps?

(8) In setting up a pump with steam-thrown valves, what precautions should be taken in order to have the valves work properly?

(9) What precautions should be taken in designing the suction pipe?

(10) (*a*) What is a foot-valve and (*b*) what is its purpose?

(11) Why is it a good plan to use a relief valve on a suction pipe that is fitted with a foot-valve?

(12) In what ways may sand and grit held in suspension in the suction water be removed before entering the pump?

(13) What is the object of placing a check-valve in the delivery pipe near the pump?

(14) Explain how a water-end by-pass can aid in starting a compound steam pump.

(15) Why is it that a pump will sometimes refuse to start when air is in the pump chamber and the full pressure is on the delivery valves?

(16) Mention some of the causes of loss in efficiency of steam pumps.

(17) Describe the manner of removing the dirt and grit from the piping and cylinder and valve seats of a new steam pump.

(18) In starting a new direct-acting steam pump with dash relief valves, what precaution should be taken with regard to the relief valves?

(19) Mention some of the causes of trouble with the suction end.

(20) (*a*) In pumping hot water, if the pump works with a jerky action, what is the trouble? (*b*) How may this jerky action be stopped?

(21) If the pump pounds at the beginning of the stroke when running fast, what is probably the trouble with the suction end?

(22) (*a*) What is the effect of too little lost motion between the valve stem and valve of a duplex pump? (*b*) What is the effect of too much lost motion?

(23) How may leaks in the suction pipe be detected?

(24) How can small pinholes in the delivery pipe be stopped up?

(25) How are air chambers usually tested for leaks?

(26) Will an air chamber aid in preventing surging in long delivery pipes?

(27) Why is surging in the suction pipes not so serious as in the delivery pipes?

(28) In pumping a mixture of water and air, how may the air be removed from the water before reaching the pump?

(29) Describe the method of setting the valves of a duplex steam pump.

PUMPS.

(PART 3.)

EXAMINATION QUESTIONS.

(1) How is the mean effective area of a double-acting piston pump found?

(2) Explain how the actual discharge of a pump may be greater than the piston displacement.

(3) What is the probable horsepower required to discharge 1,200 cubic feet of water per hour against a pressure of 125 pounds per square inch? Ans. 15.56 H. P.

(4) What is meant by the piston speed of a pump?

(5) If a pump is to discharge 94,000 pounds of water per hour, what should be the diameter of the pump plunger, its speed being 85 feet per minute? Ans. 8.22 in., nearly.

(6) (*a*) Mention three ways in which the duty of a steam pump may be stated and (*b*) compare their relative merits.

(7) The plunger of a single double-acting pump is 24 inches in diameter and the plunger rod is $3\frac{1}{2}$ inches in diameter. The plunger makes 35 strokes per minute, the length of stroke being 36 inches. What is the displacement in Winchester gallons per hour?

Ans. 146,478 gal. per hr.

(8) A single-acting plunger pump has a plunger 10 inches in diameter and a stroke of 30 inches. If the pump makes

40 discharging strokes per minute and discharges 48.3 cubic feet of water, what is the slip? Ans. 11.4 per cent.

(9) If a single double-acting pump has plungers 8 inches in diameter and makes 35 strokes per minute, how many Winchester gallons may the pump be expected to deliver per minute the length of stroke being 30 inches?

Ans. 182.9 gal. per min.

(10) If a pump requires 20 pounds of coal to raise 975 cubic feet of water 140 feet, what is the duty of the pump per 100 pounds of coal? Ans. 42,656,250 ft.-lb.

(11) Explain how the increased size of pipes and passages of large pumps increases the efficiency of the pumps.

(12) Approximately, how many cubic feet of water per minute can a 40-horsepower engine discharge at a height of 96 feet? Ans. 154 cu. ft.

(13) As usually stated, how does the efficiency of a rotary or centrifugal pump differ from the efficiency of a reciprocating steam pump?

(14) If it is desired to pump water against a pressure of 350 pounds per square inch, what should be the minimum diameter of the steam piston for a pump having a plunger 9 inches in diameter, the available steam pressure being 90 pounds per square inch? Ans. 21.1 in.

(15) (*a*) Mention some of the merits of rotary pumps. (*b*) For what class of work are they particularly adapted?

(16) About what horsepower will be required to discharge 48 cubic feet of water per minute, the total lift being 188 feet? Ans. 24.4 H. P.

(17) What is one serious objection to the use of steam driven crank-and-flywheel pumps for boiler feeding?

(18) A double-acting pump has a plunger 26 inches in diameter and 44 inches stroke. The plunger has a piston rod 4 inches in diameter extending through both pump cylinder heads. During a 12-hour duty trial the total heat supplied in the steam to the engine was 188,765,300 B. T. U. and the pump made 64,800 strokes. If the average pressure

indicated by a gauge on the discharge pipe was 160 pounds, the average vacuum indicated by a gauge on the suction pipe 10 inches, and the difference in level between the centers of the vacuum gauge and the pressure gauge 12 feet, what was the duty of the pump? Ans. 110,996,971 ft.-lb.

(19) If the water piston of a pump is 6 inches in diameter and moves at a speed of 85 feet per minute, what will be the velocity of the water in the delivery pipe if the latter is $2\frac{1}{2}$ inches in actual diameter? Ans. 489 ft. per min.

(20) Estimate the pressure against which a 35-horsepower pump can discharge 62.5 cubic feet of water per minute.

Ans. 90 lb., nearly.

(21) Why is it necessary to make the water end of a fly-wheel pump heavier than the water end of a direct-acting pump?

(22) Approximately, at what height will a pump driven by a 25-horsepower engine discharge 180 cubic feet of water per minute? Ans. 51.3 ft.

(23) Find the areas of the suction and discharge pipes for a duplex double-acting pump that is to discharge 66,000 cubic feet of water per hour.

Ans. { Suction pipe 792 sq. in.
Delivery pipe 316.8 sq. in.

(24) Estimate the number of Winchester gallons of water a pump of 75 horsepower will deliver per minute against a pressure of 150 pounds per square inch.

Ans. 602 gal. per min.

(25) If a pump lifted 128,000 cubic feet of water 85 feet with 7,280 pounds of steam, what was the duty per 1,000 pounds of dry steam? Ans. 93,406,593 ft.-lb.

(26) Roughly estimate the discharge in gallons of a double-acting pump with an 8-inch plunger.

Ans. 208.64 gal. per min.

(27) What is the principal difference between general-service pumps and tank or light-service pumps?

(28) As generally constructed, what is the distinguishing feature of pressure pumps with regard to the plunger arrangement?

(29) If the piston speed is 96 feet and the number of delivery strokes 48 per minute, what should be the length of the stroke? Ans. 2 ft.

(30) Mention some of the distinguishing features of the modern high-pressure mine pumps.

(31) How are the valves of sewage pumps frequently made so as to allow large objects to pass through the pump?

(32) Mention one advantage of the electrically driven pump when used for mining purposes.

(33) What do you understand by dry and wet vacuum pumps?

(34) (*a*) In compound and triple-expansion direct-acting duplex pumps, what is the usual ratio of expansion obtained? (*b*) How is this ratio sometimes increased?

(35) To what is the high duty of the crank-and-flywheel pump due?

A KEY TO ALL THE QUESTIONS AND EXAMPLES

CONTAINED IN THE EXAMINATION QUESTIONS INCLUDED IN THIS VOLUME

The Keys that follow have been divided into sections corresponding to the Examination Questions to which they refer, and have been given corresponding section numbers. The answers and solutions have been numbered to correspond with the questions. When the answer to a question involves a repetition of statements given in the Instruction Paper, the reader has been referred to a numbered article, the reading of which will enable him to answer the question himself.

To be of the greatest benefit, the Keys should be used sparingly. They should be used much in the same manner as a pupil would go to a teacher for instruction with regard to answering some example he was unable to solve. If used in this manner, the Keys will be of great help and assistance to the student, and will be a source of encouragement to him in studying the various papers composing the course.

THE STEAM ENGINE.

(1) The head end of a cylinder is the end farthest away from the crank-shaft; the crank end is the end nearest the crank-shaft. See Art. **4.**

(2) The distance passed over by the piston in moving from one extreme position in the cylinder to the other. See Art. **4.**

(3) The cylinder with its heads and steam chest, the steam pipe and exhaust pipe, the guides, the bed, and the bearings. See Art. **6.**

(4) The mechanism by which the steam is distributed, when considered as a whole, is termed the valve gear. See Art. **6.**

(5) The linear distance between the piston and the cylinder head when the former is at the end of its stroke is the piston clearance. The volume of this space added to the volume of the steam port leading to it is the clearance volume. See Art. **7.**

(6) The diameter of the circle described by the center of the eccentric is generally defined as the throw, but some people consider the eccentricity as the throw. See Art. **12.**

(7) By outside lap is meant the amount that the outside edge of the valve projects beyond the edge of the steam port when in mid-position; by inside lap is meant the amount that the inside edge of the valve projects beyond the edge of the steam port when in mid-position. See Art. **15.**

(8) No. See Art. **16.**

(9) The angle the eccentric radius makes with the position it would occupy if the valve had neither outside lap nor lead. See Art. **20.**

(10) Increase the outside lap of the valve, increase the valve travel, and the angle of advance. See Arts. **29, 60, 62,** and **63.**

(11) The exhaust port will be opened later and closed earlier. See Art. **30.**

(12) Lead is the distance the steam port is opened when the piston is at the end of its stroke. See Art. **31.**

(13) Ahead, irrespective of whether the engine runs under or over. See Art. **32.**

(14) By using a rocker-arm. See Art. **34.**

(15) No. The eccentric must be placed behind the crank. See Art. **35.**

(16) The exhaust port will be closed later on the forward stroke than on the return stroke, and hence there will be more compression at the head end. See Art. **41.**

(17) To double the port opening. See Arts. **42** and **43.**

(18) See Art. **46.**

(19) The volume swept through by the piston is $14^2 \times .7854 \times 28 = 4{,}310$ cubic inches. Then, the clearance is $\frac{247 \times 100}{4{,}310} = 5.73$ per cent. Ans. See Art. **51.**

(20) The apparent cut-off is $\frac{21 \times 100}{60} = 35$ per cent. Applying rule **1**, Art. **55,** we find the real cut-off to be $\frac{(35 + 2) \times 100}{100 + 2} = 36.27$ per cent. Ans.

(21) The ratio of expansion, by Art. **56,** is $\frac{100}{36.27} = 2.76$. Ans.

(22) Since the lap circle is smaller, without having changed the position of its center, the point *d* of Fig. 25 of

the text will be farther to the right, and, consequently, the perpendicular dropped from it will also be farther to the right, thus showing that cut-off will take place later. Since the circumference of the lap circle is at a greater distance than before from the line *a b*, the lead line *g h* is also farther away from it, which shows that the lead will be greater. The points of release and exhaust closure will not be affected at all, but the port opening will be greater. In case the port was opened originally its full width, the port opening obviously cannot be greater now than the width of the port; in this case it shows that the valve overtravels, i. e., moves in the same direction in which it moved to open the port after the port is wide open. The amount of overtravel will be equal to the difference in the width of the port and the distance of the circumference of the lap circle from the center *o*. See Arts. **60, 61,** and **62.**

(23) The piston speed is $\frac{80}{12} \times 75 \times 2 = 1{,}000$ feet per minute. The area of the piston is $44^2 \times .7854 = 1{,}520$ square inches, nearly. Applying rule **2**, Art. **66,** we get

$$c = \frac{1{,}520 \times 1{,}000}{6{,}000} = 253.3 \text{ square inches}$$

as the area of the steam pipe. The corresponding diameter is $\sqrt{\frac{253.3}{.7854}} = 18$ inches, nearly. By applying rule **3**, Art. **67,** we get

$$b = \frac{1{,}520 \times 1{,}000}{4{,}000} = 380 \text{ square inches}$$

as the area of the exhaust pipe. The corresponding diameter is $\sqrt{\frac{380}{.7854}} = 22$ in., nearly. Ans.

(24) The piston speed is 1,000 feet per minute. Applying rule **4**, Art. **68,** we get

$$a = \frac{1{,}520 \times 1{,}000}{7{,}500} = 202.7 \text{ sq. in. Ans.}$$

THE INDICATOR.

(1) The pressure per square inch required to raise the pencil 1 inch on the paper drum. See Art. **3.**

(2) About equal to one-half the boiler pressure. See Art. **3.**

(3) A reducing motion is a mechanism used to communicate the motion of the crosshead, on a reduced scale, to the paper drum of an indicator. See Art. **10.**

(4) Their joints are apt to become loose and thus distort the diagram. See Art. **21.**

(5) By Art. **19,** $UV = \frac{40 \times 3}{24} = 5$ in. Ans.

(6) See that it is clean and in working order. The joints of the various links should be free but not slack enough to allow the pencil to shake. Also, see that there is no backlash between the piston and spring. See Art. **27.**

(7) The vacuum line is the line of no pressure and is located below and parallel to the atmospheric line at a distance equal to the atmospheric pressure divided by the scale of the spring used to make the diagram. See Art. **31.**

(8) Corliss diagrams have the events of the stroke sharply defined and the expansion and compression curves are comparatively even. Diagrams from high-speed engines do not show the events so distinctly and the expansion and compression lines are usually irregular. See Art. **34.**

(9) (*a*) Decrease the angular advance of the eccentric. See Art. **36.**

(*b*) Cut-off, release, and compression will be later. See Art. **36.**

(10) Change the length of the valve stem so as to make the cut-off occur earlier on the end doing the greater amount of work. See Art. **44.**

(11) (*a*) The higher the speed, the more compression is needed. See Art. **49.**

(*b*) $\frac{9}{10}$ the initial pressure with high-speed engines, $\frac{6}{10}$ with medium-speed engines, and from $\frac{2}{10}$ to $\frac{3}{10}$ with slow-speed engines. See Art. **49.**

(12) (*a*) There is probably some restriction in the steam passage leading from the boiler to the cylinder. See Art. **56.**

(*b*) The free escape of the exhaust is prevented. See Art. **57.**

(*c*) The valve leaks and allows steam to enter after cut-off. See Art. **53.**

(13) (*a*) Admission, release, and compression are all too late. The back pressure is also excessive.

(*b*) Shift the eccentric ahead on the shaft and make the exhaust port or exhaust pipe larger.

(14) The rounding of the corners at the beginning of the stroke indicates that admission is a trifle late, otherwise the diagrams are very good. See Art. **37.**

(15) On account of the reevaporation of some of the condensed steam near the end of the stroke. See Art. **52.**

(16) By prolonging the expansion line and noting where it leaves the actual line of the diagram. See Art. **54.**

(17) (*a*) To the vibrations of the pencil motion when there is a sudden change of pressure. See Art. **58.**

(*b*) To the sticking of the indicator piston. See Art. **59.**

(18) Compression is too early on the head end, as is shown by the compression line extending above the steam line. There is also an unequal distribution of power in the two ends of the cylinder, as shown by the cut-off being much later on the crank-end diagram, which shows that the valve gear has been displaced. Release is too late on the crank-end diagram. See Arts. **42** and **44.**

(19) The valve on the engine from which Fig. 20 was taken had no lead, while that from which Fig. 22 was taken had lead. See Art. **42.**

(20) No. The steam might leak out as fast as it leaks in. See Art. **53.**

(21) Admission, cut-off, and release are too late on the crank end of the cylinder, thus giving an excess of power above the head end.

ENGINE TESTING.

(1) (*a*) When a gas expands and does work and no heat is added to it from an outside source the expansion is said to be adiabatic. See Art. **7.**

(*b*) When the temperature of a gas is kept constant during expansion, the expansion is said to be isothermal. See Art. **8.**

(2) The work is the same. See Art. **11.**

(3) If dry, saturated steam expands adiabatically, some of it will be condensed, and the relation between pressure and volume will then depend on the proportion of water present in the mixture. If, on the other hand, the expansion is isothermal, the steam will be superheated. See Art. **17.**

(4) (*a*) By rule **1**, Art. **20**, the work is

$W = 144\,PV = 144 \times 45.6 \times 6.3 = 41{,}368.32$ ft.-lb. Ans.

(*b*) The number of foot-pounds per minute is $41{,}368.32 \times 76$, and the horsepower developed is $\frac{41{,}368.32 \times 76}{33{,}000} = 95.27$ H. P. Ans.

(5) (*a*) The mean ordinate is the ordinate whose length is the average of all the ordinates of the diagram. See Art. **24.**

(*b*) It is found by dividing the area of the diagram by its length. In case the area is not known, divide the length of the diagram into a number of equal parts, and half way between these parts or points of division draw vertical lines

extending from the upper to the lower lines of the diagram. To find the mean ordinate, add the lengths of these ordinates or vertical lines and divide by their number. See Arts. **24** and **26.** It may also be found by a planimeter.

(6) (*a*) Volume of cylinder = area × length = 1 × 2 = 2 cubic feet. The horizontal scale of volumes

$$= \frac{\text{volume of cylinder in cubic feet}}{\text{length of card in inches}} = \tfrac{2}{3} \text{ cu. ft. per in.} \quad \text{Ans.}$$

See Art. **21.**

(*b*) Work per stroke of piston = area × horizontal scale × vertical scale × 144 or work = $6.75 \times \frac{2}{3} \times 50 \times 144 = 32{,}400$ ft.-lb. Ans. See Art. **26.**

(7) By taking the difference between the indicated horsepower when the engine is running loaded and the indicated horsepower when it is running light. See Art. **32.**

(8) The approximate net H. P. = I. H. P. − friction H. P. = 176.8 − 25.6 = 151.2. The efficiency by rule **2**

$$= \frac{151.2}{176.8} = .855 = 85.5 \text{ per cent.} \quad \text{Ans.}$$

See Art. **33.**

(9) $\frac{2.75}{3.15} \times 50 = 43.65$ lb. per sq. in., M. E. P. Ans.

See Art. **36.**

(10) By passing around the diagram two or three times and noting the reading at the return to the starting point, each time. The difference between the readings should be the same each time or the last reading divided by the number of readings should equal the first reading. See Art. **41.**

(11) By subtracting the sum of the lengths of the ordinates of the loops from the sum of the lengths of the ordinates of the main part of the diagram and dividing by the number of ordinates. See Art. **45.**

(12) Using rule 3 and the table, Art. **46,** 75 + 14.7 = 89.7. From the table the constant for $\frac{2}{3}$ cut-off is .927, and

.927 × absolute pressure = .927 × 89.7 = 83.15. M. E. P. = (83.15 − 17) × .9 = 59.54 lb. per sq. in. Ans.

(13) (*a*) Piston speed is the distance traveled by the piston in 1 minute.

(*b*) By rule **5**, Art. **48**, $S = \frac{24 \times 180}{6} = 720$ ft. per min. Ans.

(14) It is a number obtained by combining into a single factor all the constant horsepower factors for that engine. See Art. **50.**

(15) (*a*) The length L of the stroke is $\frac{24}{12} = 2$ feet; the area A of the piston is $18^2 \times .7854 = 254.47$ square inches, and the number of strokes N is $2 \times 185 = 370$. Substituting these values in the formula corresponding to rule **6**, we have

$$C_n = \frac{\frac{24}{12} \times 254.47 \times 185 \times 2}{33{,}000} = 5.706. \quad \text{Ans.}$$

(*b*) Multiplying the engine constant by the M. E. P., we have I. H. P. = 5.706 × 53.8 = 306.98. Ans. See Art. **51.**

(16) (*a*) Scale of spring $s = 60$; length L of stroke is $\frac{36}{12} = 3$ feet; area A of piston $= 24^2 \times .7854 = 452.39$ square inches, and the number N of working strokes is $2 \times 150 = 300$ per minute. The number n of ordinates is 20. Substituting these values in rule **10**, we have

$$C_o = \frac{60 \times 3 \times 452.39 \times 300}{33{,}000 \times 20} = 37.013. \quad \text{Ans.}$$

(*b*) The I. H. P. = 37.013 × 16 = 592.2. Ans. See Art. **56.**

(17) The horsepower measured by some type of absorption dynamometer. See Art. **60.**

(18) The net pressure on the scale = 274.5 − 14.5 = 260 pounds. Substituting, in rule **11**, Art. **61**, we have

$$\text{H. P.} = \frac{260 \times 4 \times 225 \times 6.2832}{33{,}000} = 44.55. \quad \text{Ans.}$$

(19) The lever arm in this case is $\frac{\frac{47}{2}+\frac{1}{2}}{12} = 2$ feet and the net pull is $241 - 10 = 231$ pounds. Substituting in rule **11**, Art. **61**, we have

$$\text{H. P.} = \frac{231 \times 2 \times 350 \times 6.2832}{33,000} = 30.78. \quad \text{Ans.}$$

(20) When fresh steam enters the cylinder, part of it is condensed and is not taken account of by the indicator diagram. See Art. **65.**

(21) By calculating the water consumption at cut-off and then at release. See Art. **67.**

(22) Length of stroke $= \frac{20}{12} = 1\frac{2}{3}$ feet; hence, each inch of length of diagram equals $\frac{1\frac{2}{3}}{3.25} = .51$ foot of stroke. As a 45-pound spring was used to make the diagram, $a\,m = .70 \times 45 = 31.5$ pounds and $b\,n = .62 \times 45 = 27.9$ pounds. Also, $O\,m = 3.41 \times .51 = 1.74$ feet and $O\,n = .36 \times .51 = .18$ foot. Area of piston $= \frac{16^2 \times .7854}{144} = 1.396$ square feet. The volume of steam in the cylinder when the piston is at the point represented by a is $1.74 \times 1.396 = 2.429$ cubic feet. The volume when the piston is at b is $.18 \times 1.396 = .25128$ cubic foot. The weight of a cubic foot of steam at an absolute pressure of 31.5 pounds is .07768 pound, and at a pressure of 27.9 pounds the weight is .06931 pound. The weight of steam in the cylinder is $.07768 \times 2.429 = .18868$ pound and the weight saved by compression is $.06931 \times .25128 = .017416$ pound. The steam used per stroke is $.18868 - .017416 = .17127$ pound. See Art. **66.**

The M. E. P. $= \frac{2.41 \times 45}{3.25} = 33.37$ pounds. See Art. **36.**

The I. H. P. is $\frac{33.37 \times 1\frac{2}{3} \times 16^2 \times .7854 \times 160 \times 2}{33,000} = 108.4.$ See Art. **47,** rule **4.** The water consumption per I. H. P. per hour is $\frac{.17127 \times 160 \times 2 \times 60}{108.4} = 30.34$ lb. See Art. **66.**

(23) The pressure at a is $50 \times .71 = 35.5$ pounds, absolute. The weight of a cubic foot of steam at this pressure is .086916 pound. Using rule **12,** we have

$$Q = \frac{13,750 \times 2.93 \times .086916}{37.5 \times 3.36} = 27.79 \text{ lb. per I. H. P. per hr.}$$

See Art. **68.** Ans.

VALVE GEARS.

(PART 1.)

(1) Slide valves, rotary valves, and poppet valves. See Art. **8.**

(2) To give a greater range of cut-off without affecting the points of admission, release, and compression. See Art. **9.**

(3) The practical range of cut-off is very small, and the space in the main-valve chamber is filled with steam at cut-off which destroys some of the benefits of early cut-off by increasing the clearance during the period of expansion. See Art. **11.**

(4) The eccentric controlling the main valve is set 90° + the angle of advance ahead of the crank, the same as for a plain slide valve. The auxiliary eccentric is set nearly opposite the crank. See Art. **13.**

(5) After setting the main valve so that the lead is the same at both ends, turn the engine to the point at which it is desired to have cut-off take place; then turn the cut-off eccentric, until the edge of the cut-off valve just closes the passage of the main valve that is open to the cylinder; then fasten the eccentric. See Art. **15.**

(6) The plain **D** slide valve requires considerable force to drive it, which not only absorbs considerable of the power of the engine, but greatly affects the action of the governor. When used with a governor which changes the throw of the eccentric. the port opening is greatly restricted at early

cut-offs, which produces wiredrawing. If the valve is given overtravel to overcome this latter difficulty, the valve has to be made larger and requires still more power to move it.

(7) In horizontal engines the wear due to the weight of the valve does not affect the tightness of the valve, and if the valve does wear so that it leaks, it can usually be made tight without renewing any of the parts. The pressure-plate valve also can be made so as to act as a relief valve for the escape of water from the cylinder. See Arts. **20** and **21.**

(8) The steam should be admitted to the steam chest slowly so as to warm the valve and its seat gradually and thus prevent the valve from binding in its seat. See Art. **21.**

(9) Not very seriously. The space back of the valve is connected with the exhaust port so that the pressure in this space is always the same as the pressure in the exhaust port. See Art. **22.**

(10) To quickly close the steam valve after it has been disconnected from the cut-off mechanism. See Art. **29.**

(11) The steam valves are so arranged that, when they are moving at their most rapid rate, steam is being admitted to the cylinder, their period of slow motion corresponding to the time during which they are closed. In a similar manner the most rapid motions of the exhaust valves take place at release and compression. See Arts. **31** and **32.**

(12) The valve will work as though positively connected to the admission crank. See Arts. **34** and **40.**

(13) Suppose the valves do not have lap or lead and the releasing gear does not work, then the steam valve closes at the extreme end of the stroke and the corresponding exhaust valve opens at the same instant, thus giving the steam a chance to blow through. This occurs on both ends of the stroke. It is necessary, then, to give the steam valve sufficient lap to close it before the exhaust valve opens and

prevents its opening until the exhaust valve has closed. In order to have prompt release and sufficient compression to check the motion of the reciprocating parts of the engine, the exhaust valves must be given a certain amount of lead. See Art. **40.**

(14) When a single eccentric is used, it is necessary to give it a considerable angle of advance in order to give a proper steam distribution. As the releasing mechanism cannot act later than $\frac{1}{2}$ stroke, this angle of advance greatly reduces the effective range of cut-off. By the use of two eccentrics, one for the exhaust valves and one for the steam valves, the range through which the governor can control the cut-off is greatly extended, as the cut-off eccentric can be set 90°, or even a little less, ahead of the crank. See Art. **49.**

(15) The steam valves will open very slowly at the beginning of the stroke, thus producing wiredrawing of the steam. See Art. **50.**

(16) The so-called high-speed Corliss engine has the advantage of the four oscillating valves and wristplate motion of the ordinary Corliss engine, and as its valves are positively driven, they can operate at a much higher speed. The degree of regulation is also superior because the wristplate is driven by a shifting eccentric controlled by a shaft governor. By the use of two eccentrics all the other advantages of the ordinary Corliss gear, using two eccentrics, may be obtained. See Art. **53.**

(17) By changing the length of the carrier rod, so as to move the wristplate towards its center of motion, by an amount equal to $\frac{1}{2}$ the overtravel. See Art. **43.**

(18) Cut-off is made earlier; the other events remain the same. See Art. **13.**

(19) Overtravel is the amount the valve travels over that which is necessary to give a full port opening. See Art. **18.**

(20) To secure an increased port opening with a small valve travel. See Art. **18.**

(21) Shorten the eccentric rod until the valve has the same lead on both ends; then decrease the angle of advance of the eccentric until the required lead is obtained. See Art. **15.**

VALVE GEARS.

(PART 2.)

---

(1) No. To vary the cut-off it is necessary to use another eccentric and a separate cut-off valve. See Art. **5.**

(2) In the Stephenson link motion the link is shifted and the block remains stationary, while in the Gooch link motion the block is shifted and the link remains stationary. See Arts. **6** and **7.**

(3) The eccentrics should always point towards the link, whether the crank is on its inner or outer dead center. See Art. **11.**

(4) Yes; there is always a slight port opening when the link is in mid-gear. See Art. **12.**

(5) The lead remains constant. See Art. **10.**

(6) By changing the length of the eccentric rod. See Art. **13.**

(7) As the lead with a Stephenson link changes for difent positions of the link, it is sometimes found that when the valve is set with the link in full gear, the lead is not satisfactory for the running position. Hence, it is better to give the valve proper lead when the link is in the running position. See Art. **13.**

(8) Because of the force required to hold them and to move them to a new position. They require a very powerful governor, whose action in most cases is too slow to secure close regulation. See Art. **17.**

(9) No; the sum of the two leads is a constant quantity that cannot be readily changed. See Art. **21.**

(10) The single-seat poppet valve is easily made and kept tight, but is difficult to balance. The double-seat poppet valve is easily balanced to any extent, but is difficult to keep tight on account of unequal expansion in the valves, the stem, or seat. See Art. **23.**

(11) (*a*) The steam distribution can be made almost perfect because the cams can be made of such shape that the valves will be opened and closed rapidly at any point of the stroke. See Art. **26.**

(*b*) At high speeds the gear is noisy and its action is unsatisfactory. See Art. **26.**

(12) A large port opening is obtained with a small valve travel, which greatly reduces the wear on the valve and seat. The power required to move the valve is also reduced. See Art. **30.**

(13) (*a*) No; the cut-off will generally be different for both ends of the cylinder. See Art. **42.**

(*b*) By giving the eccentric rod an angle with the rocker that will affect the valve motion in the same way that the angle of the connecting-rod affects the motion of the piston. See Art. **42.**

(14) The increased lead secures an earlier admission of steam to the lower end of the cylinder, which aids in overcoming the weight of the reciprocating parts. See Art. **43.**

(15) First prepare the tram and remove the steam-chest cover, then set the valve so that it just covers one of the steam ports. On some convenient fixed part of the engine make a prick-punch mark; place one point of the tram in this mark; and where the other point reaches the valve stem make another prick-punch mark. Now move the valve so that the other port is just covered and make a similar mark. By the aid of a tram and these two marks, the valve can be set without removing the steam-chest cover. See Art. **47.**

(16) Put the crank on one center and place the gear in full-gear position. Place the valve in the position corresponding to the crank position and give it a little lead. Place the crank on the opposite center and measure the lead. Then equalize the lead by shifting the valve one-half the difference between the two leads towards the port having the greater lead. See Art. **21.**

(17) With the Joy valve gear, lead and compression are not greatly increased with early cut-off, while with the Stephenson link motion with open rods, they both become considerably greater. See Art. **18.**

(18) No; one port is always opened farther than the other. See Art. **19.**

(19) (*a*) The lead is increased with open rods and (*b*) decreased with crossed rods. See Art. **12.**

(20) The togglejoint is so arranged that its parts are taking or leaving their angular position when the valve is opening or closing, thus giving the valve a very rapid motion during these periods. During the time the valve is closed its motion is very slow, owing to the joint being near its straight-line position. The motion of the valve is thus similar to the motion of a Corliss valve. See Art. **32.**

CONDENSERS.

(1) On account of the air contained in the exhaust steam and the air that leaks in around the piston rod and valve stems, and also on account of the vapor from the water. See Art. **9.**

(2) To remove the air, vapor, and condensed steam and sometimes, also, the condensing water from the condenser. See Art. **11.**

(3) (*a*) In a jet condenser the exhaust steam is condensed by mingling with the condensing water. See Art. **15.**

(*b*) In a surface condenser the exhaust steam is condensed by coming into contact with metallic surfaces that are kept cool by water constantly flowing over them. See Art. **16.**

(4) Water is apt to be drawn into the cylinder and cause the blowing out of a cylinder head. See Art. **21.**

(5) When the condenser is deprived of injection water. See Art. **21.**

(6) A cubic inch of water converted into steam at atmospheric pressure occupies 1,646 cubic inches; therefore, if 1,646 cubic inches of steam contained in a closed vessel at atmospheric pressure are condensed into water, a vacuum will be formed. Thus, by making the engine exhaust into

a closed vessel and condensing the exhaust steam, the pressure on the exhaust side of the piston is made less and consequently the net pressure on the piston will be increased. See Arts. **9** and **11.**

(7) By means of an indicator card. See Art. **12.**

(8) By compounding the steam cylinders of the independent air and circulating pumps; or, in case of multiple-expansion engines, by running the exhaust of the independent engines into the receiver of the low-pressure cylinder of the main engine. See Art. **13.**

(9) By a column of water flowing downwards through a vertical pipe not less than 34 feet long and having its lower end immersed in the water of the hotwell. See Art. **29.**

(10) To increase the velocity of the falling water, which improves the action of the condenser. See Art. **30.**

(11) (*a*) No. If there is a vacuum of 24½ inches, the pump is required to force the water but 7 feet. See Art. **31.**

(*b*) Because the vacuum assists the circulating pump. See Art. **31.**

(12) (*a*) A surface condenser. See Art. **40.**

(*b*) Because the exhaust steam does not come into direct contact with the impure injection water. See Art. **40.**

(13) (*a*) Two. See Art. **41.**

(*b*) The circulating pump forces the injection water through the condenser tubes and the air pump removes the air, vapor, and water of condensation from the condenser. See Art. **41.**

(14) The surface condenser is more complicated, costs more, and requires more attention than the jet condenser, but the pure feedwater obtained more than compensates for these disadvantages. See Art. **42.**

(15) It relieves the condenser of any excess of steam, air, or vapor that may accumulate within it, and it allows

the engine to be run non-condensing should the air pump become inoperative. See Art. **43.**

(16) Because the water in the boiler will soon become as impure as the injection water and it will then be necessary to blow off some of the very impure water and to replace it with less impure water, thus causing a serious loss of heat. See Art. **45.**

(17) (*a*) The coating of the tubes by the grease carried from the cylinder by the exhaust steam. See Art. **46.**

(*b*) If animal or vegetable oils are used, the condenser may be cleaned by boiling it out with a solution of caustic soda or caustic potash. If mineral oils are used, the tubes must be removed and the grease scraped from them by hand. See Art. **46.**

(*c*) By introducing a grease extractor in the exhaust pipe. See Art. **48.**

(18) (*a*) Copper, 70 per cent.; zinc, 29 per cent.; and tin. 1 per cent. See Art. **50.**

(*b*) They should be tinned both inside and outside. See Art. **50.**

(19) (*a*) By a screw packing gland that prevents leakage between the tube and tube-sheet and yet allows the tube to freely change its length. See Art. **51.**

(*b*) Because if they were rigidly fastened to the tube-sheets, they would become greatly distorted through unequal expansion and contraction. See Art. **52.**

(20) By removing the condenser bonnets and filling the steam side of the condenser with water. See Art. **53.**

(21) By a falling vacuum. See Art. **54.**

(22) To prevent the formation of a galvanic current between the copper of the tubes and the iron of the condenser casing, feedpipes, and boilers. See Art. **55.**

(23) On the principle that the evaporation of a part of the water undergoing the cooling process extracts the heat from the remaining part. See Art. **57.**

(24) The amount of surface exposed, the condition of the air with regard to the moisture it contains, the temperature of the air, and the amount of air brought in contact with the water. See Arts. **59** and **60.**

(25) To evaporate a pound of water at the atmospheric pressure requires 966 B. T. U., and as this amount of heat must come from the condensing water, the remaining water must be cooled. See Art. **61.**

(26) The cooling tower consists of a tower about 30 feet high, to the top of which the water to be cooled is delivered. As the water descends from the top of the tower, it meets an ascending current of air and also with obstructions so placed that it falls in a fine spray or thin sheets, thus exposing a large area of evaporating surface to be acted upon by the air. See Art. **62.**

(27) (*a*) By radiation, by contact with the cool air, and by evaporation. See Art. **63.**

(*b*) Evaporation. See Art. **63.**

(28) The Linde system of cooling discharge water consists of a number of horizontal thin metallic cylinders immersed to one-third their diameters in the condensing water. By revolving the cylinders a thin film of water, which adheres to the surface of the cylinders, is brought into contact with a current of air and thus produces a cooling effect. See Art. **71.**

(29) By allowing a certain amount of air to enter the barrel of the circulating pump at each stroke, or by means of a regurgitating valve. See Arts. **75** and **76.**

(30) By the use of a vacuum breaker. See Art. **82.**

(31) On the initial and final temperature of the steam and on the initial and final temperature of the condensing water. See Art. **83.**

(32) From the Steam Tables we find H to be 1,131.462. Applying rule **1,** we get

$$W = \frac{1{,}131.462 - (135 - 32)}{105 - 55} = 20.57 \text{ lb., nearly. Ans.}$$

(33) Not enough condensing water is being used. See Art. **87.**

(34) From the Steam Tables, H for 3 pounds absolute is 1,125.144. Applying rule **1**, we get

$$W = \frac{1{,}125.144 - (130 - 32)}{130 - 65} = 15.8 \text{ lb.} \quad \text{Ans.}$$

(35) About 2 inches less than the theoretical vacuum. See Art. **88.**

COMPOUND ENGINES.

(1) The pressures, and consequently the strains, are less, and are more evenly distributed throughout the stroke; the turning moment on the shaft is more uniform, which allows lighter parts to be used; the flywheel can also be lighter, and thus there is less friction. See Art. **9.**

(2) On account of initial cylinder condensation. See Art. **3.**

(3) By expanding the steam successively in several cylinders the range of temperature in each cylinder is reduced, as is also the fluctuation in temperature of the cylinder walls of each. As the rate of heat transmission is proportionally much smaller for a small than for a large fluctuation of temperature, the sum of the condensation losses in the several cylinders will be smaller than the condensation losses in a single cylinder having the same temperature range between the initial and final temperature. See Art. **7.**

(4) The greater part of the condensed steam in the high-pressure cylinder is probably reevaporated during the exhaust period and enters the low-pressure cylinder as steam, and thus does some work. See Art. **8.**

(5) (*a*) A tandem compound engine. See Art. **14.**

(*b*) Because the turning moment of the tandem compound engine is not as uniform as that of the cross-compound engine. See Art. **14.**

(6) In the Woolf compound type the pistons commence and complete the stroke at the same time. The pistons may operate upon one crank or upon two or more cranks that are either in line or placed 180° apart. In the receiver compound type the high-pressure cylinder exhausts into a separate vessel in order to allow the cranks to be placed at any desired angle other than 0° or 180° apart. See Arts. **16** and **17.**

(7) Assuming that cut-off occurs at the same point in both cylinders, the high-pressure exhaust will be compressed by the high-pressure piston after cut-off occurs because the exhaust has no place to go to. This compression represents a waste of work, and to overcome it a receiver must be used. See Art. **18.**

(8) To the number of different stages in which the steam is expanded. See Art. **19.**

(9) The weights of the two sets of reciprocating parts balance each other. There are no dead centers and the turning effort is the same as if two cranks set nearly 90° apart were used. Also, the stresses on the crankpins are gradually changed around the pins and not suddenly reversed, as is the case with an ordinary engine. See Art. **27.**

(10) By drop is meant the difference in pressure at release and in the receiver. See Art. **29.**

(11) Owing to the free expansion of the steam, the drop in a receiver tends to superheat the steam and thus make it drier. See Art. **30.**

(12) (*a*) Making the high-pressure cut-off later increases the receiver pressure; making the cut-off earlier decreases it.

(*b*) If the low-pressure cut-off remains the same, the volume of steam removed from the receiver per stroke remains constant regardless of the amount of steam entering the receiver. But as the high-pressure cut-off is changed, the amount of steam entering the receiver is changed, and thus the pressure must vary, owing to a constant volume being taken from it. See Art. **31.**

(13) By changing the cut-off in the low-pressure cylinder. See Art. **33.**

(14) Less work will be done in the high-pressure cylinder and more in the low-pressure cylinder. See Art. **34.**

(15) When the high-pressure cut-off occurs late, the low-pressure cylinder does the greater share of the work, and when the cut-off occurs very early, the high-pressure does the greater part of the work, and in some cases the low-pressure cylinder may then act as a drag on the engine. See Art. **35.**

(16) To relieve it of undue pressure, which would result should the low-pressure cut-off be set too early or should the low-pressure admission valves fail to open when a releasing gear is used. See Art. **37.**

(17) (*a*) No. See Art. **39.**

(*b*) Because the initial load is entirely independent of the relative amount of work done in the two cylinders. See Art. **39.**

(18) So as to obtain the highest economy in conjunction with a satisfactory mechanical operation. See Art. **42.**

(19) (*a*) To prevent cylinder condensation. See Art. **44.**

(*b*) No. See Art. **44.**

(*c*) Because if the steam is sufficiently superheated there will be no cylinder condensation. See Art. **44.**

(20) The low-pressure valves should have the most lead. See Art. **38.**

(21) (*a*) A reheater consists essentially of a receiver containing a nest of pipes through which high-pressure steam circulates, and around which the working steam must circulate before entering the cylinder. See Art. **47.**

(*b*) To thoroughly reheat the steam before it enters the cylinder in which it does its work. See Art. **47.**

(22) It indicates that the reheater is wasting heat. See Art. **47.**

(23) Because the steam will then be admitted to both sides of the reheater as soon as the engine is started, and thus its expansion will be more uniform. See Art. **49.**

(24) (*a*) To the ratio between the volume of steam in the low-pressure cylinder, at low-pressure release, and the volume of steam admitted to the high-pressure cylinder. See Art. **51.**

(*b*) No. See Art. **52.**

(25) By rule **2**, we have

$$E = \frac{12,460 \times 3}{3,115} = 12. \quad \text{Ans.}$$

(26) As a ratio of expansion by volume, or as a ratio of expansion by pressure. See Arts. **52** and **53.**

(27) By rule **1**, we have

$$E = \tfrac{8800}{800} = 11. \quad \text{Ans.}$$

(28) The initial absolute pressure is approximately $(180 + 14.7) - 5 = 189.7$, say 190 pounds per square inch. The terminal pressure may be taken as 9 pounds (see Art. **59**). The ratio of expansion by pressure is $\tfrac{190}{9} = 21.1$, say 21. By Table I, the factor for a ratio of expansion of 21 is .192. The back pressure, by Art. **59,** may be estimated at 3 pounds. The factor to be taken from Table II is .60.

Applying rule **3**, we get

$$p_m = (190 \times .192 - 3) \times .60 = 20.03,$$

say 20 pounds per square inch.

The probable indicated horsepower is

$$\frac{20 \times \frac{24}{12} \times 39^2 \times .7854 \times 2 \times 180}{33,000} = 521.2, \text{ say } 521. \quad \text{Ans.}$$

ENGINE MANAGEMENT.

(PART 1.)

---

(1) When the steam enters the cold pipe or cylinder, it condenses and forms a partial vacuum that causes a still larger volume of steam to enter the pipe or cylinder. This action is repeated until a mass of water collects, which will rush through the steam pipe with the steam, and on striking a bend or other obstruction cause water hammer. See Art. **19.**

(2) The valve should be eased on its seat to prevent its getting stuck by the unequal expansion of the valve casing. See Art. **20.**

(3) All oil should be wiped from the bright and painted parts before it has time to set; rusty spots on the bright work should be removed. All oil holes not fitted with oil cups should be carefully plugged and any dirt about the engine or engine room should be removed. See Arts. **22** and **23.**

(4) First, thoroughly warm the steam pipe by slightly opening the stop-valves and allowing the steam to flow until it issues from the drain cock near the throttle; then close the drain cock and slightly open the throttle or the by-pass around the throttle, thus allowing the steam to enter the steam chest and cylinder. The drain cocks on the cylinder and steam chest should also be opened, and if the cylinders are steam-jacketed, steam should be turned into the jackets

and the jacket drain cocks opened. To further warm the cylinder, steam should be admitted to both ends by means of a by-pass valve or by turning the engine over by hand. See Art. **17.**

(5) By allowing the hot air coming from the boilers before steam is generated to circulate through the cylinder and steam pipe. See Art. **18.**

(6) From the upper or lower half center. See Art. **24.**

(7) To jump the crank over the first center, after which the flywheel will carry it over the other centers. See Art. **25.**

(8) There is probably water in the cylinder. See Art. **27.**

(9) If the drain cocks are closed, the steam will condense inside the engine and thus allow it to cool off slowly, which will lessen the danger of cracking the cylinder. See Art. **29.**

(10) By simply throwing the link from one side to the other. See Art. **30.**

(11) In order to allow the amount of circulating water to be readily adjusted to suit variations in the temperature of the water and to suit the degree of vacuum desired. See Arts. **33** and **35.**

(12) The cylinder and steam pipe should first be warmed up, and just previous to starting the engine the injection and delivery valves of the condenser should be opened and the air and circulating pumps started. The engine may then be started by simply opening the throttle. After the engine has been running a few minutes, the speed of the air and circulation pumps and the admission of injection water should be regulated. See Arts. **34** and **35.**

(13) (*a*) If the injection valve is not opened at the moment the engine is started, the condenser will be filled with air and steam, which will prevent the injection water entering. See Art. **36.**

(*b*) When the condenser gets hot, it is necessary to pump cold water into it or to cool it by playing cold water upon it before it can be started. See Art. **36.**

(14) (*a*) The snifting valve is used on jet condensers. It is similar to a safety valve, but is held to its seat simply by its own weight and the pressure of the atmosphere. See Art. **37.**

(*b*) Its purpose is to relieve the pressure on the inside of the condenser when it gets hot. See Art. **37.**

(15) The eccentric rod is unhooked from the wristplate and a starting bar inserted in the wristplate. The throttle is then opened and the wristplate vibrated back and forth, by hand, by means of the starting bar. In this manner the steam and exhaust valves are operated and the engine started. After the engine has made several revolutions, the eccentric rod is hooked onto the wristplate and the starting bar unshipped. See Art. **43.**

(16) The eccentric rod is unhooked and the Corliss engine stopped in the desired position by the operation of the wristplate by hand. The slide-valve engine is stopped in the desired position by operating the throttle. See Art. **48.**

(17) If pass-over or starting valves are provided they may be opened, or the steam may be worked into the low-pressure cylinder by operating the high-pressure valves by hand. See Art. **49.**

(18) If the pressure is too high or too low in the receiver, the engine will not start. See Art. **50.**

(19) (*a*) Reversible compound engines. See Art. **52.**

(*b*) Because they can be started much easier when there is a vacuum in the condenser. See Art. **52.**

(20) It should be set at the greatest cut-off in order that the engine may be started promptly. See Art. **56.**

(21) They should be parallel to each other and perpendicular to the center line of the connecting-rod. See Art. **62.**

(22) Secure a strip of wood across the head end of the cylinder by means of the stud bolts and through the strip of wood bore a 1-inch hole approximately in line with the center of the cylinder. At the crank end of the bed erect a standard with a hole similarly located. Through these holes tightly stretch a fine cord, fastening the cord in such manner that it can be shifted about in the holes. Now locate the cord central with the head end of the cylinder by shifting it about until it measures the same distance from all sides of the cylinder. In a similar manner locate the cord central with the crank end of the cylinder. The location of the cord should then be verified for both ends of the cylinder. See Arts. **64, 65,** and **66.**

(23) By dropping a plumb-line from overhead and touching the line to be leveled and then shifting the line until it is perpendicular to the plumb-line. See Art. **73.**

(24) (*a*) The center line of the shaft is usually left a little higher than the center line of the cylinder. See Art. **75.**

(*b*) So that when the brasses and journals wear to a bearing, the center line of the shaft will be nearly level with the center line of the cylinder. See Art. **75.**

(25) The journals are given a coat of red or black marking material and the shaft is placed in position and rocked back and forth, the lower brasses alone being in position. The shaft is then removed and the high spots of the brasses scraped down. This operation is repeated until the desired bearing surface is obtained; then the upper brasses are fitted in a similar manner. See Art. **79.**

(26) It may be done by turning the shaft until the crankpin touches the line stretched through the center of the cylinder and measuring the distance from the line to one of the shoulders on the crankpin. Then turn the shaft so that the crankpin touches the line on the other side of the center of the shaft, and if the line is the same distance from the shoulder of the crankpin as before, the shaft is set correctly. See Art. **80.**

(27) Place the crank on one dead center, connect the connecting-rod to the crankpin, leaving it free at the crosshead end, and key up the brasses snugly; then measure the distance from some point on the crosshead end of the connecting-rod to one of the crosshead guides. Place the crank on the opposite center and repeat the measurement. If both measurements are alike, the crankpin is parallel to the shaft in the horizontal plane. The same operation should then be repeated with the crankpin at the upper and lower half centers. See Art. **83.**

(28) The crankpin may be given a thin coat of Prussian blue or red-lead paint and the rod then connected snugly to the wristpin, but not so snugly to the crankpin. After turning the crank through one revolution, examine the crankpin brasses, and if the coloring matter has spread evenly over them, their correct adjustment may be assumed. See Art. **86.**

(29) Loose journal brasses. See Art. **89.**

(30) By reducing the two halves of the brasses so that a number of thin strips of brass may be placed between them. One or more of these strips of brass may then be removed when the boxes become brass bound. Or, keepers of cast brass, cast iron, or wood may be placed between the boxes instead of the brass strips. See Arts. **90** and **91.**

(31) Usually there is but little space between the piston-rod nut and the cylinder head, so that the former cannot back off very far before it will strike and break the cylinder head. See Art. **95.**

(32) If there is too little compression, the piston rod, connecting-rod, and crank-shaft will be suddenly thrown forwards at the dead centers, where the direction of pressure is suddenly reversed, thus causing pounding, as there is always some lost motion at the pin and shaft bearings. See Art. **101.**

(33) (*a*) Yes. See Art. **105.**

(*b*) Pressure is retained so long on the driving side of the piston that there will not be sufficient compression to stop the piston gradually. See Art. **105.**

(34) By allowing sufficient air to enter the barrel of the pump to form a cushion for the piston. See Art. **112.**

ENGINE MANAGEMENT.

(PART 2.)

---

(1) (*a*) If it were not for the wearing of the bearings and journals, the solid bearing would be the ideal bearing, as it cannot be tampered with by careless persons. Its main disadvantage is that it cannot be adjusted, which is necessary on account of wear. See Arts. **3** and **4.**

(*b*) The advantage of adjustable bearings is that the wear may be taken up as fast as it occurs, but through carelessness or ignorance they are liable to be taken up too much and thus cause the bearing to heat. See Art. **4.**

(2) If increasing the oil feed does not stop the heating, mix some flake graphite, flour sulphur, or powdered soapstone with the oil and feed the mixture into the bearing. Aqua ammonia will also sometimes stop the heating. See Art. **6.**

(3) The brasses are liable to warp or crack by unequal contraction. See Art. **8.**

(4) Stop the engine and slack off the brasses; then keep the inside of the bearing deluged with a mixture of oil and graphite, sulphur, or soapstone. If necessary, cold water may be applied to the outside of the bearing. See Art. **11.**

(5) By wearing down a bearing is meant the running of the bearing until the rough rubbing surfaces of the brasses and journal have become smooth and are in their normal working condition. See Art. **15.**

(6) It is difficult to replace the brasses exactly as they were before removal; consequently the brasses do not bear evenly on the journal, which causes heating. See Art. **16.**

(7) (*a*) Yes. See Art. **20.**

(*b*) By the hammering of the journal against the brasses. See Art. **20.**

(8) The brasses may be set up solid on the journal and then slacked off until, by trial, it is found that they work properly, or thin strips of metal may be put between the brasses and the bearing set up tight. Enough strips should be used to cause the brasses to set loosely on the journal; then by removing a pair of strips at a time the brasses can be set up to the proper point. See Arts. **21** and **22.**

(9) If not too bad, the brasses may be filed, scraped, or chipped to fit the journal, but if too greatly distorted, they will have to be replaced by new ones. See Art. **23.**

(10) Badly fitting brasses. See Art. **25.**

(11) (*a*) When the brasses are heated quickly, they tend to expand along the surface in contact with the journal; which would tend to make the bore of larger diameter; but this expansion is prevented by the cooler portions of the brasses and by the outer part of the bearing. The layer of metal near the journal thus receives a permanent set, so that the brasses close on the journal when they become cooler. See Art. **26.**

(*b*) By chipping off the brasses at their thin edges parallel to the journal. See Art. **27.**

(12) All oil should be strained or filtered before it is used and all oil cups, oil cans, and oil channels should be frequently cleaned. See Art. **30.**

(13) (*a*) It should have a high degree of viscosity. See Art. **33.**

(*b*) Because oil having a high degree of viscosity offers more resistance to being squeezed from between the journal and brasses than does thin oil. See Art. **33.**

(14) By chipping oil channels in the brasses. See Art. **33.**

(15) On the materials of which the brasses and journals are composed, the fineness of their finish, the accuracy of their fit, the adjustment of the brasses, and the lubricant used. See Art. **38.**

(16) Overloading an engine increases the pressure on the journals. This increased pressure may cause the practical limit of pressure allowable on the bearings to be exceeded and thus cause the bearings to heat. See Art. **41.**

(17) Because it promotes a better distribution of oil and prevents the journal and brasses wearing into concentric parallel grooves. See Art. **48.**

(18) On the amount of greasy particles that it contains. See Art. **52.**

(19) It should reduce friction to a minimum. It should be free from acids, alkalies, and disagreeable odors. It should not be altered by exposure to the air and should stand a low temperature without solidifying or depositing solid matter. It should be free from grit and all foreign matter. Cost is also a consideration. See Art. **52.**

(20) (*a*) No.

(*b*) Because the specific gravity of an oil is not an indication of its viscosity, neither is the viscosity of an oil an indication of its specific gravity. See Art. **53.**

(21) (*a*) Pure lard oil. See Art. **57.**

(*b*) Olive oil. See Art. **58.**

(22) (*a*) Mineral oils are distilled from bituminous shale and from the residuum of crude petroleum after the volatile oils and illuminating oils have been distilled off.

(*b*) According to their specific gravity. See Art. **59.**

(23) Because they are liable to have a low flashing point. See Art. **61.**

(24) Boiled, or cup, greases are made by saponifying fats and fatty oils with lime and dissolving the soap in mineral oil. See Art. **67.**

(25) The lubricant spreads itself in the form of a thin film between the rubbing surfaces and thus prevents their coming in direct contact with one another. See Art. **75.**

(26) Because they are not carbonized by the high temperature nor are they decomposed by heat, as are animal and vegetable oils, which break up into acids that attack the metal of the pistons, valves, etc. See Art. **76.**

(27) The higher the temperature of the bearing, the less is the lubricating power of the oil. See Art. **78.**

(28) By rubbing the greases between the forefinger and thumb or in the palm of the hand. See Art. **83.**

(29) The point or temperature at which the vapor rising from the oil will ignite with a flash. See Art. **84.**

(30) Boil about a pint of the oil, into which there has been placed 1 or 2 ounces of caustic soda or concentrated lye, for $\frac{1}{2}$ hour and allow it to cool. If, when cool, the surface of the oil is covered with soap, the oil contains animal or vegetable fats; otherwise it is pure mineral oil. See Art. **86.**

(31) The viscosity is increased. See Art. **87.**

(32) Sulphur. See Art. **89.**

(33) (*a*) Steam is allowed to enter the reservoir containing the oil and as it condenses, the water, being heavier than the oil, sinks to the bottom, thus raising the level of the oil until it overflows into a suitable passage. See Art. **92.**

(*b*) The flow of oil cannot be readily controlled and there is no means of telling when the lubricator stops working. See Art. **102.**

(34) (*a*) No.

(*b*) Because on closing the throttle there will be full steam pressure on the condenser and none on the sight-feed

glass. In consequence, the lubricator will be rapidly emptied. See Art. **111.**

(35) (*a*) Mechanical, water-displacement, and hydrostatic. See Art. **92.**

(*b*) On the pressure of the head of water furnished by condensation of steam. See Art. **92.**

(36) In a double-connection lubricator the steam is admitted through one pipe and the oil leaves through the other pipe. In a single-connection lubricator the oil passes through the same pipe that admits the steam. See Art. **104.**

ENGINE INSTALLATION.

(1) (*a*) The available floor space and the size of the engine. See Art. **2.**

(*b*) The first reason is self-evident. As to the second reason, the piston of large horizontal engines, and particularly the low-pressure piston of compound engines, becomes so massive that it is almost impossible to support it in such manner that it will not seriously wear the lower side of the cylinder. See Art. **2.**

(2) Accessibility for repairs, inspection, and oiling, lower cost, the cylinders are easily drained, and in general it requires much less physical strength to properly take care of it. See Arts. **4, 5, 6,** and **7.**

(3) (*a*) Yes.

(*b*) They are of the four-valve type, in which the cut-off valve is mounted on the back of the main valve and is positively driven by a shaft governor. See Art. **12.**

(4) Yes; the throttling engine with a Meyer valve by which the cut-off can be varied from 0 to $\frac{3}{4}$ stroke is fairly efficient. See Art. **16.**

(5) The cost of fuel and the amount of power required. See Art. **17.**

(6) The compound engine is more complicated and hence more liable to break down, which requires more spare parts to be carried. In order to keep it in proper working order, a higher degree of skill is required and the boilers required

are more expensive, although there may not be as many of them. The facilities for repairs should also be considered. See Art. **21.**

(7) The inaccessibility of the cylinders and pistons for inspection or repairs. See Art. **23.**

(8) The friction of the crossheads and crankpins is the same for both classes of engines and the friction should also be the same for the pistons and piston rods of both engines, but owing to wear the friction is slightly greater in the tandem compound. The friction of the valve gear is in favor of the tandem compound, but the tandem compound requires a flywheel $1\frac{6}{10}$ times heavier than the cross-compound; this requires a heavier shaft and larger bearings, which greatly increases the frictional resistance. See Art. **25.**

(9) The volume of the receiver is made about equal to the volume of the low-pressure cylinder and there are about 50 square feet of tube-reheating surface for each cubic foot of steam exhausted by the high-pressure cylinder. See Art. **28.**

(10) There is difficulty of securing sufficient compression in the low-pressure cylinder to absorb the inertia of the reciprocating parts. See Art. **29.**

(11) Their advantages are comparatively low first cost and small space required; their principal objections are the extra care required to keep them in order and their wastefulness of fuel. See Art. **36.**

(12) (*a*) Piston valves and valves balanced by cover-plates or pressure plates. See Art. **37.**

(*b*) The piston valve gives fairly good results on vertical engines, but when used on horizontal engines it is liable to leak badly after running a short time. Owing to the method of regulation, the piston valve wears very unevenly, which further causes it to leak. Valves balanced by pressure plates require less clearance, and when properly fitted wear better than balanced valves. See Art. **37.**

(13) Owing to the high rotative speed, the regulation of high-speed engines is much closer than the regulation of slow-speed engines. See Art. **40.**

(14) Owing to the high speed, but little time is allowed for initial condensation or for change of temperature in the cylinder walls between the strokes. See Art. **45.**

(15) There is a direct saving due to the omission of transmission machinery, such as jack-shafts, belts, or gearing and bearings. The cost of lubrication, attendance, and repairs is also reduced and the foundations are less expensive. See Art. **46.**

(16) The valves are designed to give a minimum clearance volume and clearance surface, the cylinders and cylinder heads are steam-jacketed, the internal surfaces of the heads and the pistons are polished to prevent initial condensation, care is taken to free the cylinders of water, and the valve gears are carefully constructed to give a theoretical steam distribution. See Art. **48.**

(17) (*a*) Their principal advantage is that they allow a much smaller armature or revolving field to be used on the generator, which greatly reduces the first cost of the unit. See Art. **50.**

(*b*) After passing 200 or 300 horsepower, the saving in first cost is more than offset by the increased economy of the slow-speed engine; hence it is not usual to use high-speed engines for large direct-connected units. See Art. **50.**

(18) A compound condensing high-speed engine having separate steam and exhaust valves and having a governor controlling the admission valve only would be most efficient under the conditions named. See Art. **54.**

(19) Because the economical range of cut-off is much greater for the compound condensing engine than for the compound non-condensing engine. See Art. **56.**

(20) Because it can be readily converted into a duplex engine, a condensing engine, or a compound condensing

engine when the increased power required demands it. See Art. **57.**

(21) In general, if fuel is very cheap, a cheap simple engine is selected, but if fuel is dear, then a more expensive engine of the multiple-expansion type is selected in order to secure as small a steam consumption as possible. See Arts. **67** and **68.**

(22) No, probably not. Superheating and jacketing both tend to reduce initial cylinder condensation, so where one is used there is nothing gained by using the other also. See Art. **68.**

(23) The cooling tower as usually constructed consists of a round or rectangular tower so arranged that when the water from the condenser is delivered at the top of the tower, it is divided into a great number of fine sprays in its descent. An artificial current of air cools the water as it falls and thus renders it fit for injection again. See Art. **78.**

(24) In many manufacturing plants the various departments require widely varying speeds or are required to run overtime or all night, or they may require power for but a short time; under these conditions the subdivided power will give the best results. See Art. **79.**

(25) By suspending a heavy mass underneath the floor, but rigidly bolted to the engine base. See Art. **83.**

(26) (*a*) Engine foundations are usually made of brick, dressed stone, or concrete. See Art. **86.**

(*b*) If brick or stone are used, they should be laid in Portland cement mortar, as lime mortar disintegrates under vibration. See Art. **86.**

(27) (*a*) The vibrations of the engine will be transmitted directly to the rock and hence to adjoining property. See Art. **83.**

(*b*) The rock is usually covered with a layer of timber or rubble or layer of sand 2 or 3 feet deep; the foundation is then built on this footing. See Art. **88.**

(28) The foundation is usually supported on piles driven from $2\frac{1}{2}$ to 4 feet apart from center to center. To form a footing for the foundation a timber grating is fastened to the top of the piles and a layer of concrete is deposited, or the space between the piles is filled with rubble, clay, or concrete. The foundation proper is then built upon the footing thus prepared. See Art. **87.**

(29) If the capstone is too low, the bearing can be easily shimmed up to the proper height, but if it is too high, it must be reset or chipped down to the proper height. See Art. **90.**

(30) By means of a templet constructed of wood, on which have been carefully marked the center lines of the engine and crank-shaft and the relative position of the bolt holes. The templet is suspended in its correct position above the foundation and the bolts are passed through holes bored in the templet. The bolts are located at the proper height by blocks placed on the templet, the height of the blocks depending on the height of the engine bosses. See Art. **95.**

(31) The space is usually filled with grouting, which may be made of iron borings mixed with cement, sal ammoniac, sulphur, and water, or melted sulphur or pure Portland cement may be used alone. See Art. **99.**

PUMPS.

(PART 1.)

---

(1) On account of mechanical imperfections, air contained in the water, and the vapor of the water itself. See Art. **6.**

(2) Because the atmospheric pressure at the bottom of the mine is greater than at the surface. See Art. **7.**

(3) Because the increased vapor pressure at the higher temperatures counteracts the pressure of the atmosphere. See Art. **8.**

(4) A steam pump in which the pressure of the steam in the steam cylinder is transferred to the pump piston or plunger in a straight line. See Art. **14.**

(5) The difference between the power exerted in the steam cylinders and the resistance in the pump. See Art. **14.**

(6) The intermittent motion of the column of water being pumped. See Art. **15.**

(7) The steam valves of each pump are driven by the piston rod of the opposite pump. See Art. **16.**

(8) (*a*) Because it is necessary to carry the full steam pressure the full length of the stroke. See Art. **18.**

(*b*) By compounding the steam end, or by the use of a high-duty attachment, or both. See Art. **18.**

(9) There is an exhaust port and steam port for each end of the cylinder. The exhaust port is some distance from the end of the cylinder, so that the piston covers it before the end

of the stroke, thus confining some steam in the cylinder to act as a cushion. See Art. **25.**

(10) The dash relief valves control a passage between the steam and exhaust ports and are so set that at the highest speed of the pump there will be sufficient compression to prevent the piston striking the cylinder heads. As the speed becomes slower, the compression remaining the same, the piston would stop short of the full stroke, if it were not that the dash relief valves allow the compressed steam to escape through the steam port, thus allowing the piston to complete its stroke. See Art. **28.**

(11) (*a*) To keep a more uniform pressure in the steam chests of the low-pressure cylinders. See Art. **34.**

(*b*) The steam chests of the low-pressure cylinders are joined by a pipe that allows the exhaust from the high-pressure cylinders to pass to either of the low-pressure cylinders. When the pressure begins to drop in one low-pressure steam chest, the pressure is highest in the opposite low-pressure steam chest, and as the two steam chests are connected, the steam pressure is nearly equalized in both. See Art. **34.**

(12) It allows the steam to be cut off early in the cylinders, thus allowing the steam to be used expansively. See Art. **36.**

(13) (*a*) The high-duty attachment, as usually made, consists of two compensating cylinders having their plungers attached to opposite sides of the pump crosshead. These cylinders are connected to an accumulator through hollow trunnions, on which they oscillate as the pump crosshead moves backwards and forwards. At the beginning of the stroke the plungers are forced into the compensating cylinders, thus creating a pressure in the accumulator, but after the pump crosshead has passed the center of its stroke the angle between the compensating plungers and the pump piston rod becomes such that the plungers are forced out and thus aid in completing the stroke. See Arts. **36** and **37.**

(*b*) The results obtained are independent of the speed. See Art. **38.**

(14) It allows the steam pistons to work at a higher speed, which is a decided advantage in many respects. See Art. **44.**

(15) The Quimby pump has two shafts placed side by side and connected by gears. Each shaft has a right-handed and left-handed screw, the right-handed screw of one shaft meshing with the left-handed screw of the other shaft. The screws fit the casing closely and are a close running fit on each other. The water passes through passages in the casing to the water ends of the screws and is then drawn towards the center by the revolving screws and is discharged through the discharge pipe. See Art. **48.**

(16) (*a*) On the pressure produced by the centrifugal force of a quantity of water rotated by the vanes of the pump. See Art. **49.**

(*b*) They are particularly adapted to low heads where large quantities of water are to be pumped and also where water containing large quantities of mud, sand, and gravel is to be handled. See Art. **50.**

(17) 90° apart. See Art. **53.**

(18) Because they get their power with the same efficiency as the engine from which they are driven. See Art. **56.**

(19) Because the leakage can be easily stopped and the plunger type is best adapted to high pressures. See Art. **58.**

(20) A pit pump is a pump having its water end located at the bottom of the mine and connected to a steam engine or other motor at the surface. See Art. **59.**

(21) The momentum of the pit work is sufficient to complete the stroke after the steam is cut off; this allows the steam to be used expansively. See Art. **61.**

(22) The heavy walking beam and its connections are dispensed with, the first cost is less, there is less friction, and the advantage of a direct-acting engine is obtained. See Art. **63.**

(23) The pump rod reduces the effective area of the pipe and increases the friction of the water. The rods are concealed and cannot be readily inspected. When the rods or bolts break, it is difficult to recover them. When pumping against a heavy pressure, it is impossible to keep the piston tight. See Art. **67.**

(24) The suction pipe is called the wind bore and the delivery pipe the working barrel. See Art. **68.**

(25) When putting down a new shaft or deepening an old one, the pump used to drain the water from the shaft bottom is called a sinking pump. See Art. **71.**

(26) It has no wearing parts except the valves, which are easily and cheaply repaired. It will work in any position and requires no attention when once started. There are no parts to get out of order and it will pump anything that can get past the valves. See Art. **86.**

(27) It has no moving parts and is not affected by sand or grit. The action of the air purifies the water and cools it as it is being pumped. It is also claimed to increase the flow of a well. The full area of the well is available for a flow of water. See Art. **88.**

(28) The differential pump has two pistons whose displacements are in the ratio of 2 to 1. The larger piston works in the suction chamber and the smaller piston in the delivery chamber. When the large piston enters the suction chamber, it forces into the delivery chamber a volume of water equal to its displacement, but at the same time the small plunger has withdrawn from the delivery chamber a volume equal to half the large plunger displacement. Thus the amount of water actually discharged is equal to the difference in the displacement of the two plungers, or equal to

the displacement of the small plunger. On the return stroke the small plunger discharges an amount equal to its displacement and at the same time double the amount of water is drawn into the suction chamber. See Arts. **94** and **95.**

(29) The suction valve, which is positively seated just before the end of the suction stroke by a buffer on the water plunger. See Art. **102.**

PUMPS.

(PART 2.)

---

(1) (*a*) With hemp contained in a stuffingbox of the ordinary pattern. See Art. **3.**

(*b*) With a **U**-shaped leather packing held in a recess in the upper end of the pump cylinder. See Art. **4.**

(2) When the packing becomes worn, the heads of the pump cylinder must be removed to tighten or renew it, and there is no way of detecting leakage when the pump is working. See Art. **5.**

(3) The part of the pump chamber that contains the valves. See Art. **14.**

(4) To give the valve a partial rotation at each stroke of the pump so that it will wear its seat evenly. See Art. **20.**

(5) To relieve the pipes and pump of shocks by promoting a uniform flow of water. See Art. **23.**

(6) (*a*) An alleviator is simply a plunger working in a cylinder through a stuffingbox; the plunger is forced into the cylinder by springs or rubber buffers. The cylinder communicates with the delivery pipe of the pump and thus acts the same as an air chamber. See Art. **28.**

(*b*) When pumps work against high pressures, the air in the air chambers is rapidly absorbed by the water or escapes from the air chamber, thus rendering it useless. For this reason alleviators are used. See Art. **28.**

(7) To promote a prompt flow of the water into the pump chamber. See Art. **29.**

(8) The foundation surface should not be winding nor should the steam or water pipes be sprung into place, else the valves will be liable to stick. See Art. **39.**

(9) The suction pipe should be as straight as possible, and if bends are necessary they should be of large radius. It should be of one size from end to end, and if very long it should be somewhat larger than is necessary to keep the velocity of flow down to 200 feet per minute. If the lift is high, a suction chamber and a foot-valve should be provided. See Art. **41.**

(10) (*a*) A foot-valve is a check-valve placed at the lower end of the suction pipe and opening towards the pump. See Art. **42.**

(*b*) Its purpose is to prevent the suction pipe emptying while the pump is standing and to prevent the water in the suction pipe slipping back while running. See Art. **42.**

(11) Because if the suction valves leak or if the priming pipe is left open, the full pressure of the delivery water will come on the suction pipe, which is usually not designed to withstand such a high pressure. See Art. **43.**

(12) By means of a settling chamber, a suction basket, a strainer, or a special form of strainer consisting of perforated plates placed in the suction pipe near the pump cylinder. See Arts. **44** and **45.**

(13) To relieve the pump of pressure when starting up, so that it will take hold of the water more readily, and to hold back the water in case of repairs. See Art. **46.**

(14) By opening the by-pass the pressure on the plungers can be relieved for a sufficient number of strokes to allow steam to enter the low-pressure cylinder, thus rendering the full power of the steam end available for pumping. See Art. **48.**

(15) The air is not dislodged but only compressed and expanded again with the motion of the piston; thus no vacuum is formed and the pump will not start. See Art. **52.**

(16) Wear, improper adjustments, wrong timing of the movements of the steam valve, leakage, lack of alinement, and foreign matter in the suction and foot-valves and suction and delivery pipes. See Art. **58.**

(17) The pistons, valves, and cylinder heads are removed, and as the steam pressure rises in the boiler it is allowed to blow through, thus thoroughly cleaning the piping. Before the working pressure is reached, the stop-valves are closed and the cylinder heads put on and the stuffingboxes closed, leaving the pistons and valves still out of the cylinders. The steam at full working pressure is then turned on, which thoroughly removes all dirt and grit from the valve seats and cylinders. Any dirt found in the corners of the cylinders should then be removed by hand. See Arts. **63** and **64.**

(18) The dash relief valves should be closed in order to keep the piston as far from the cylinder heads as possible. See Art. **72.**

(19) Leaks at the joints or along the suction pipe or in the pump chamber, which may be caused by imperfect connections, leaky chaplets, shifted cores, blowholes, corrosion, or cracks from frost. See Art. **76.**

(20) (*a*) The lift is too high for the temperature of the water. See Art. **77.**

(*b*) By decreasing either the lift or the temperature. See Art. **77.**

(21) The pump chamber is not filling and the plunger is striking the incoming water on its return stroke. See Art. **78.**

(22) (*a*) Short stroking. See Art. **83.**

(*b*) The piston will strike the heads. See Art. **83.**

(23) By the ear, by the flame of a candle, or by stopping the lower end of the suction pipe and putting a pressure of 40 or 50 pounds per square inch on it. See Art. **85.**

(24) By spreading a thick layer of red-lead putty over the leaks and then wrapping several layers of canvas covered with red-lead putty on both sides tightly about the pipe. See Art. **88.**

(25) By closing all openings and then pumping air into them until the working pressure is reached. If the chambers are tight, the air pressure should show no reduction in 24 hours. See Art. **90.**

(26) No; they probably aggravate the trouble by forming a cushion from which the column of water rebounds. See Art. **97.**

(27) Because the direction of the force resulting from surging in the suction pipe is in the natural direction of flow of the water and simply tends to open the pump valves, while the shock due to surging in the delivery pipe comes against the valves and must be withstood by the machinery. See Art. **99.**

(28) By connecting an air pump to the suction air chamber. See Art. **102.**

(29) Move the pistons until they strike the cylinder heads and make a mark on the piston rod at the end of the steam-end stuffingbox gland. Move the pistons until they strike the opposite cylinder heads and make another mark on the piston rod. Then make a mark half way between these two marks and move the pistons until these central marks come even with the end of the stuffingbox gland. Now set the valves central over the ports and adjust the locknuts so as to allow the same lost motion on each side of the valve. See Art. **103.**

PUMPS.

(PART 3.)

---

(1) The mean effective area is equal to twice the piston area diminished by the area of the piston rod, and the difference divided by 2. See Art. **4.**

(2) When the suction and discharge pipes are long and the lift moderate, there may be sufficient energy imparted to the column of water during the discharge stroke to keep it in motion during the return stroke. Under these conditions the actual discharge may be larger than the displacement. See Art. **8.**

(3) Reducing the volume per hour to pounds per minute, we have

$$\frac{1,200 \times 62.5}{60} = 1,250 \text{ pounds per minute.}$$

Applying rule **4,** Art. **15,** we have

$$H_e = \frac{1,250 \times 125}{10,043} = 15.56 \text{ H. P.} \quad \text{Ans.}$$

(4) The number of feet traveled per minute by the plunger when discharging water. See Art. **20.**

(5) Reducing the weight of water discharged per hour to cubic feet discharged per minute, we have

$$\frac{94,000}{62.5 \times 60} = 25.07 \text{ cubic feet.}$$

Applying rule **9**, Art. **22**, we get

$$d = \sqrt{\frac{229 \times 25.07}{85}} = 8.22 \text{ in., nearly. Ans.}$$

(6) (*a*) The duty may be stated as the number of foot-pounds of work done per 100 pounds of coal burned, or per 1,000 pounds of dry steam, or per 1,000,000 B. T. U. supplied. See Arts. **35, 36,** and **38.**

(*b*) Duty based on the coal consumption gives an idea of the coal required by a pump of a given type for the performance of a given quantity of work, but it does not give reliable results when the duty of pumps of different types working under different conditions are to be compared. The basis of 1,000 pounds of dry steam is better adapted to duty trials, but it is open to the objection that the pressure of the steam is not taken into consideration. The heat-unit basis is the most scientific and the most accurate for comparative purposes, as the duty is based on the exact amount of heat energy consumed by the pump. See Arts. **35, 37,** and **38.**

(7) The mean effective area of the plunger is

$$\frac{24^2 \times .7854 + (24^2 \times .7854 - 3\tfrac{1}{2}^2 \times .7854)}{2}$$
$$= 447.58 \text{ square inches.}$$

Applying rule **1**, Art. **3**, we have

$$D_{ag} = \frac{36 \times 447.58 \times 35}{231} = 2{,}441.3 \text{ gal. per min.,}$$

or $$2{,}441.3 \times 60 = 146{,}478 \text{ gal. per hr. Ans.}$$

(8) By rule **1**, Art. **3**, the displacement is

$$\frac{30 \times 10^2 \times .7854 \times 40}{1{,}728} = 54.54 \text{ cubic feet.}$$

By rule **2**, Art. **8**, the slip is

$$\frac{(54.54 - 48.3) \times 100}{54.54} = 11.4 \text{ per cent. Ans.}$$

(9) The piston speed is $\frac{30}{12} \times 35 = 87.5$ feet per minute. Applying rule **10**, Art. **23**, we have

$$D = \frac{8^2 \times 87.5}{229} = 24.45 \text{ cu. ft. per min.},$$

or

$$\frac{24.45 \times 1,728}{231} = 182.9 \text{ gal. per min.} \quad \text{Ans.}$$

(10) The weight of water pumped is $975 \times 62.5 = 60,937.5$ pounds. By rule **14**, Art. **34**, we get

$$D = \frac{100 \times 60,937.5 \times 140}{20} = 42,656,250 \text{ ft.-lb.} \quad \text{Ans.}$$

(11) Increasing the size of pipes and passages reduces the relative amount of friction surface exposed per unit volume of water delivered, thus increasing the efficiency. See Art. **49**.

(12) By rule **6**, Art. **17**, we have

$$W = \frac{23,100 \times 40}{96} = 9,625 \text{ lb.},$$

or

$$\frac{9,625}{62.5} = 154 \text{ cu. ft. per min.} \quad \text{Ans.}$$

(13) The efficiency of a rotary or centrifugal pump is the efficiency of the pump itself and not of the pump and engine, as in the case of a steam pump. See. Art. **50**.

(14) The area of the plunger is $9^2 \times .7854 = 63.62$ square inches. By rule **13**, Art. **28**, we get

$$d_m = \sqrt{\frac{1.8 \times 63.62 \times 350}{90}} = 21.1 \text{ in.} \quad \text{Ans.}$$

(15) (*a*) Rotary pumps are light, simple, and inexpensive, and occupy relatively but little space for their capacity. They require but little or no foundation. See Arts. **55** and **56**.

(*b*) They are particularly adapted to pumping water holding soft material in suspension. See Art. **55**.

(16) The weight of water to be pumped per minute is $48 \times 62.5 = 3,000$ pounds. Applying rule **3**, Art. **14**, we get

$$H_e = \frac{3,000 \times 188}{23,100} = 24.4 \text{ H. P.} \quad \text{Ans.}$$

(17) They cannot always be run slow enough to suit the demand without stopping on the centers.

(18) The mean effective area of each end of the plunger is $26^2 \times .7854 - 4^2 \times .7854 = 518.36$ square inches. The pressure corresponding to a vacuum of 10 inches is $p = 10 \times .4914 = 4.91$ pounds per square inch, and the pressure corresponding to a difference in level of 12 feet is $s = 12 \times .434 = 5.21$ pounds per square inch. The stroke is $\frac{44}{12} = 3\frac{2}{3}$ feet. Applying rule **16**, Art. **44**, we get

$$D = \frac{1,000,000 \times (160 + 4.91 + 5.21) \times 518.36 \times 3\frac{2}{3} \times 64,800}{188,765,300}$$
$$= 110,996,971 \text{ ft.-lb.} \quad \text{Ans.}$$

(19) The area of the piston is $6^2 \times .7854 = 28.27$ square inches and the area of the delivery pipe is $2\frac{1}{2}^2 \times .7854 = 4.91$ square inches. By rule **18**, Art. **52**, we get

$$v = \frac{28.27 \times 85}{4.91} = 489 \text{ ft. per min.} \quad \text{Ans.}$$

(20) Reducing the cubic feet of water to be delivered to pounds, we have $62.5 \times 62.5 = 3,906.25$ pounds. Applying rule **7**, Art. **18**, we have

$$P = \frac{10,043 \times 35}{3,906.25} = 90 \text{ lb., nearly.} \quad \text{Ans.}$$

(21) The velocity of discharge of a flywheel pump is variable throughout the stroke; thus shocks are produced that make it necessary to use a heavier water end than would be used for a direct-acting pump, where the velocity of discharge is practically constant. See Art. **88.**

(22) Reducing the volume of water discharged to pounds, we have $180 \times 62.5 = 11,250$ pounds. By rule **5**, Art. **16**, we get

$$L = \frac{23{,}100 \times 25}{11{,}250} = 51.3 \text{ ft.} \quad \text{Ans.}$$

(23) Reducing the water discharged to cubic feet per minute, we have $\frac{66000}{60} = 1{,}100$ cubic feet. Applying rule **17**, Art. **51**, we get for the suction pipe

$$A = \frac{144 \times 1{,}100}{200} = 792 \text{ sq. in.,}$$

and for the delivery pipe

$$A = \frac{144 \times 1{,}100}{500} = 316.8 \text{ sq. in.} \quad \text{Ans.}$$

(24) By rule **8**, Art. **19**, we get

$$W = \frac{10{,}043 \times 75}{150} = 5{,}021.5 \text{ lb. per min.,}$$

or $$\frac{5{,}021.5}{8.34} = 602 \text{ gal. per min.} \quad \text{Ans.}$$

(25) Reducing the volume of water to pounds, we have $128{,}000 \times 62.5 = 8{,}000{,}000$ pounds. Applying rule **15**, Art. **36**, we have

$$D = \frac{1{,}000 \times 8{,}000{,}000 \times 85}{7{,}280} = 93{,}406{,}593 \text{ ft.-lb.} \quad \text{Ans.}$$

(26) By rule **11**, Art. **24**, we get

$$D_g = 3.26 \times 8^2 = 208.64 \text{ gal. per min.} \quad \text{Ans.}$$

(27) The principal difference is in the relative sizes of the steam and water cylinders, the steam cylinder being proportionally much larger for the general-service pumps than for the light-service pumps. See Art. **67.**

(28) The distinguishing feature is the four single-acting plungers working in the ends of the water cylinders. See Art. **70.**

(29) By rule **12**, Art. **26**, we have

$$L = \tfrac{96}{48} = 2 \text{ ft.} \quad \text{Ans.}$$

(30) Outside-packed plungers; strong circular valves independent of one another, but bolted to the working

chamber, to the suction and delivery pipes, and to one another. All parts are made so that they can be easily renewed and sometimes the whole water end is made of some acid-resisting bronze or is made of iron or steel and lined with some acid-resisting material. See Art. **71.**

(31) The valves are frequently made in the form of large leather-faced door or flap valves, giving the full area of the pipe. See Art. **79.**

(32) Many various sizes of pumps can be placed about the mines and driven from an economical generating unit at the surface. See Art. **82.**

(33) Dry vacuum pumps are those that handle air only, while wet vacuum pumps handle both air and water. See Art. **86.**

(34) (*a*) The ratio of expansion usually obtained is a little more than the ratio of the low-pressure cylinder to the high-pressure cylinder. See Art. **87.**

(*b*) This ratio is sometimes increased by making the reciprocating parts heavy and running the pump at some fixed minimum speed such that the inertia of the parts will complete the stroke when the steam is cut off in the high-pressure cylinder before the end of the stroke. The degree of expansion may also be increased by the use of a high-duty attachment. See Art. **87.**

(35) The high duty is mainly due to the degree of expansion that can be obtained, and also to the ease with which all the refinements necessary for high duty can be applied. See Art. **89.**

INDEX

NOTE.—In this volume, each Section is complete in itself and has a number. This number is printed at the top of every page of the Section in the headline opposite the page number, and to distinguish the Section number from the page number, the Section number is preceded by a section mark (§). In order to find a reference, glance along the inside edges of the headlines until the desired Section number is found, then along the page numbers of that Section until the desired page is found. Thus, to find the reference "Accumulator, §34, p26," turn to the Section marked §34, then to page 26 of that Section.

Q

R

T